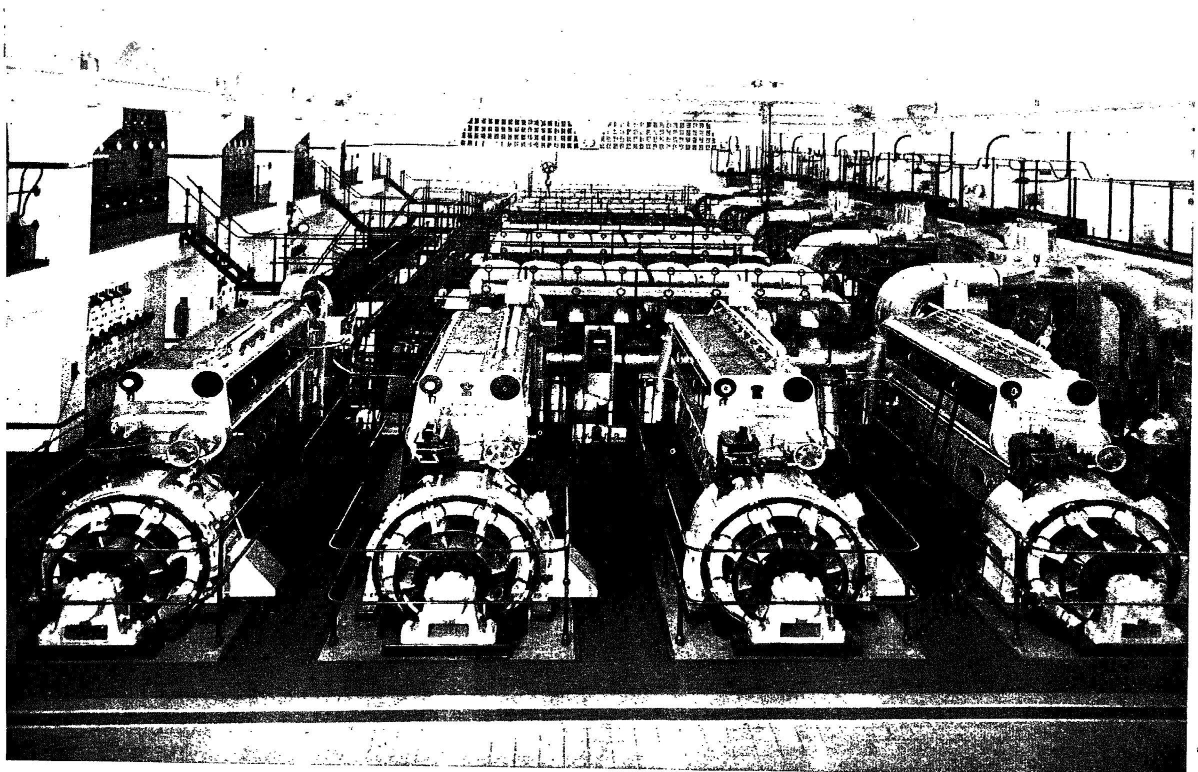

Plate I. General view of generating station, Bank of England.

HEATING AND AIR-CONDITIONING OF BUILDINGS

WITH SOME NOTES ON
COMBINED ELECTRICAL GENERATING STATIONS

BY

OSCAR FABER

O.B.E., D.C.L., D.SC., M.INST.C.E., M.I.MECH.E., P.P.I.STRUCT.E., A.M.I.E.E., F.C.G.I., M.I.H.V.E.

AND

J. R. KELL

A.M.I.MECH.E. M.I.H.V.E., M.INST.F.

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PREFACE TO SECOND EDITION

Since the publishing of the first edition numerous developments have taken place, and the present re-printing has afforded the opportunity to bring the subject matter up to date, and in line with more recent information.

For example, the Institution of Heating and Ventilating Engineers *Guide to Current Practice* has been published since the first Edition. This has undoubtedly filled a long-felt need in this country for an authoritative collection of data on heating and allied subjects.

The tables in the present book have been brought into line with the *Guide* wherever the Authors felt the newer information to be better than the old, and their thanks are due to the Code of Practice Committee for permission to reproduce such material.

The psychrometric tables published in the *Guide* differ so fundamentally from the previous basis (taken from American sources) that it has been found necessary to revise the Air-Conditioning section drastically. The psychrometric chart here included is believed to be the first attempt to present this I.H.V.E. data in graphical form, and it is hoped it will become the basis of English practice. The reader's attention is also drawn to the separate publication of a larger version of the Chart, prepared in colour, and which should be found useful for general office use.

Developments in large-scale heating systems have warranted a fuller treatment of the chapter on Steam, and a new chapter on 'Heating by High-Pressure Hot Water' has been introduced, as this method has been increasingly used for large installations.

Another addition is a section on 'Ventilation', including descriptions of the apparatus used, and charts for use in duct-sizing. The section on 'Heating by Gas' has been largely re-written, also that on combustion. Throughout the book new apparatus has been referred to as far as possible, though it must be appreciated that owing to the war much development work has no doubt been done which has not yet been published, and the post-war period may bring many further improvements. The fundamental problems will, however, always remain the same, and the student must be prepared to treat such changes, if any, as changes in method or detail rather than in principle.

Where *costs* are given it should be understood that these are based on *pre-war prices*. It is impossible to forecast what the trend of post-war costs will be, hence any reference to these figures should allow for such variations as are found to apply at the particular time when they are used. No doubt this may be done by applying a percentage addition (or reduction)

based on an investigation of the changes which have come about in the meantime.

Our thanks are due to Mr. J. R. Harrison, B.SC., D.I.C., A.C.G.I., A.M.INST.C.E., A.M.I.H.V.E., for a great deal of help in the work of revision; to Mr. G. Murray, B.SC., A.M.INST.C.E., A.M.I.MECH.E., A.M.I.GAS E., A.M.I.H.V.E., in the revision of the Gas section; and to the members of our staff for their assistance in the work of checking and of preparing illustrations. We are also indebted to the various manufacturers for permission to reproduce blocks illustrating particular equipment.

OSCAR FABER
J. R. KELL

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LIST OF PLATES

'It's snowing still,' said Eeyore gloomily.

'So it is.'

'*And* freezing.'

'Is it?'

'Yes,' said Eeyore. 'However,' he said, brightening up a little, 'we haven't had an earthquake lately.'

'What's the matter, Eeyore?'

'Nothing, Christopher Robin. Nothing important. I suppose you haven't seen a house or whatnot anywhere about?'

'What sort of a house?'

'Just a house.'

'Who lives there?'

'I do. At least I thought I did. But I suppose I don't. After all, we can't all have houses.'

'But, Eeyore, I didn't know—I always thought——'

'I don't know how it is, Christopher Robin, but what with all this snow and one thing and another, not to mention icicles and such-like, it isn't so Hot in my field about three o'clock in the morning as some people think it is. It isn't Close, if you know what I mean—not so as to be uncomfortable. It isn't stuffy. In fact, Christopher Robin,' he went on in a loud whisper, 'quite-between-ourselves-and-don't-tell-anybody, it's Cold.'

From *The House at Pooh Corner*, with grateful acknowledgements to A. A. Milne and Messrs. Methuen & Co. Ltd.

INTRODUCTION

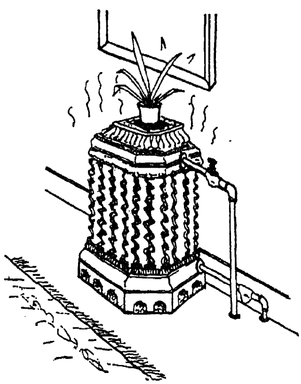
The most virile races of the world live in the temperate zone. They are subject at some times to severe cold and at others to almost tropical heat. As a result their temperature regulating apparatus has been developed to a higher degree than in more equable climates. They can survive the blizzards of the Arctic, and go campaigning in the desert or jungle.

In their own normal habitat it might be thought that they would equally by now have become acclimatized to living happily without artificial heat. Such, however, is not the case.

In winter in temperate latitudes there is a deficiency of sun, and mankind has apparently from pre-historic times sought by various means to keep himself and his house warm by fire. At first it was by a fire in the centre of the cave, in Roman times by a fire under the floor, and later by fire in a grate with a chimney to carry off the smoke. The fire became by long tradition the centre of the home, the natural focal point of all family activity, and such it will no doubt long remain.

But as buildings became larger, heating by many fires was necessary, though laborious, wasteful and dirty. Thus was evolved the system known as Central Heating, to heat many rooms from one large fire situated in the cellar or elsewhere.

Early attempts at this were not entirely successful. Central Heating was deemed to be foreign and unhealthy. And no wonder, for 'You must



'The sort of thing over which savages would have built a temple.'

remember', as Beverley Nichols has said in his recent book, *A Thatched Roof*, 'that their recollection of central heating is mainly based on a night they spent in a hotel in Worthing for Ada's wedding in November, 1902.

They occupied the royal suite, which contained the hotel's only radiator. This radiator, of Persian design, was the sort of thing over which savages would have built a temple. It hissed and gurgled and spat—at noon it boiled—at night it froze solid, and housemaids approached it on tiptoe, with nervous giggles.'

The system has long since been perfected in numerous forms to suit all kinds of purposes, but even so there are still a diminishing few who class it among the undesirable scientific novelties of the age.

The majority have, however, by now become accustomed and educated to it. The War, for instance, has brought many changes in the habits of people, one of which is the living together for the first time in huts, camps and hostels, and the working together in factories, depots and offices, all centrally heated. Those who have had this experience know the comfort of a warm building in cold weather, and will take unkindly to a return to the dampness and chilliness of clothes, furniture, tools and so on associated with a lack of proper heating. Most of our factories, public buildings, and all large establishments could indeed not function without such a system.

It may, therefore, be stated that Central Heating, which covers all means of warming from a central source, including gas and electricity, has become an essential part of building construction; just as important as the drains or electric light. The days are past when a case had to be made out for it, or when it was looked on by the Architect or Owner as a sort of poor relation, to be treated only as an afterthought at as little cost as possible.

In fact, the position is now becoming reversed. The necessity for fuel economy due to possible shortage or high cost, demands that buildings shall be easily heated. In future construction more regard must be had to insulation and weather-tightness. Planning from the start must take into account the means of heating and all the provisions necessary must be made so that economy of installation and running shall be paramount. This will call for the closest collaboration of Architect and Engineer.

If carried a stage further, centralization of heating becomes what is known as 'District Heating', in which many buildings are served from one central source. Considerable progress has been made in other countries along these lines, but it has not so far been adopted on a large scale in Great Britain. It may, however, find a place in post-war development. The subject is too great to be dealt with here, and is the subject of a separate treatise by the Authors.

The 'How' of Central Heating in its various forms, together with its allied systems, hot-water supply, ventilation and air-conditioning, constitutes the subject matter of this book. The reasons for choice of one system as compared with another are as far as possible given in each section, followed by details of the main principles of design and, where useful, some idea of costs.

The illustrations are generally of actual installations carried out under

the direction of the Authors, and are intended to help those unfamiliar with these matters to visualize the finished product.

It is intended that the presentation of the subject as a whole shall be such that all those interested in building and who feel that highly technical matters are outside their scope, may gain a clearer perception of the relative importance of the problems which have to be faced, and some idea of the ways of solving them. At the same time Engineers and Students of the subject should find the technical side carried sufficiently far to be useful in their work.

CHAPTER I

General Considerations

The function of heating and air-conditioning in buildings may be described in general as the providing of conditions of comfort for the occupants. It also embraces the problem of producing special conditions of temperature and humidity for various industrial processes, and for the storage and manufacture of a wide variety of materials.

The problem of producing conditions of comfort involves, firstly, a consideration of the type of individual who will inhabit the building and what his or her ideas of comfort will be.

Secondly, it is necessary to consider the kind of building and the method and amount of heating, ventilation or air-conditioning required to achieve the desired result.

Thirdly, the question of cost has to be examined both as regards installation and running.

The decision as to whether ventilation or air-conditioning is necessary will depend to some extent on the degree of concentration of people in the building, and its geographical location.

The dirt in the air of our modern cities, combined with the noise of traffic, often precludes the opening of windows, so that the provision by mechanical means of sufficient air at the right temperature and humidity without draughts is becoming more and more necessary.

Before these problems can be discussed it is necessary to define some of the terms which are commonly used, with special reference to the particular meaning and use which the heating engineer normally applies to them, and it will perhaps be convenient to consider these next.

DEFINITIONS

Temperature—Temperature is a measure of 'hotness', and its magnitude is referred to some well-known basic temperature such as that of boiling water.

In England the scale almost invariably used is one which was devised by a German named Fahrenheit, who placed the freezing point of water at a figure of 32 degrees and the boiling point of water at a figure 180 degrees higher, i.e. 212°. If a mercury-in-glass thermometer (which measures the relative expansion of mercury and glass) is so calibrated that the freezing point is marked 32 and the boiling point is marked 212, then any intermediate temperature is given a figure between these two, proportionate to the length of the mercury column.

The reason why he fixed the freezing point at 32 was because in his time no temperature lower than 32° below freezing point had been discovered, and he thought that the zero on his scale represented the lowest temperature which could be achieved. As it turned out, this was completely wrong, and the absolute zero, which has since been very closely

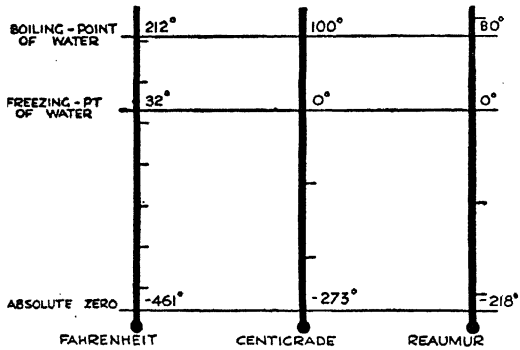


TABLE I
TEMPERATURE SCALES

approached by many experimenters, is frequently taken as −461°* Fahr. The absolute zero has been established by studying the characteristics of expansion of gases, and other phenomena, and is assumed to be the temperature of space.

In Germany, on the other hand, the scale in common use is that devised by a Frenchman named Réaumur, in which the freezing point is 0 and the boiling point 80, and intermediate temperatures are determined by the proportionate reading on a mercury column in glass.

In France and on the Continent generally, and in all scientific laboratories throughout the world, the scale of temperature in use is that devised by a Swede called Celcius, and known as the Centigrade scale, in which the freezing point of water is zero and the boiling point 100. This is by far the most useful and rational scale.

Tables I and II give equivalent temperatures on the three scales.

To convert Centigrade degrees into Fahrenheit, multiply by 9, divide by 5 and add 32, and conversely, to convert Fahrenheit into Centigrade, deduct 32, multiply by 5 and divide by 9.

It is interesting to note that the intermediate temperatures as measured by a mercury-in-glass thermometer do depend a little on the expansion characteristics of mercury and glass, and that thermometers which depend on the expansion of other materials do not necessarily give an exact subdivision between freezing and boiling point when checked against a mercury thermometer. For very accurate scientific work this has to be taken into account, but is not necessary for the purposes of this present treatise.

* Recent determinations indicate that the actual value is −459·6° F.

° F.	° C.	° F.	° C.	° F.	° C.	° F.	° C.	° F.	° C.
0	−17·78	70	21·11	120	48·89	170	76·67	220	104·44
5	−15·00	72	22·22	122	50·00	172	77·78	225	107·22
10	−12·22	74	23·33	124	51·11	174	78·89	230	110·00
15	−9·44	76	24·44	126	52·22	176	80·00	235	112·78
20	−6·67	78	25·56	128	53·33	178	81·11	240	115·56
25	−3·89	80	26·67	130	54·44	180	82·22	245	118·33
30	−1·11	82	27·78	132	55·56	182	83·33	250	121·11
32	0	84	28·89	134	56·67	184	84·44	255	123·89
34	1·11	86	30·00	136	57·78	186	85·56	260	126·67
36	2·22	88	31·11	138	58·89	188	86·67	265	129·44
38	3·33	90	32·22	140	60·00	190	87·78	270	132·22
40	4·44	92	33·33	142	61·11	192	88·89	275	135·00
42	5·56	94	34·44	144	62·22	194	90·00	280	137·78
44	6·67	96	35·56	146	63·33	196	91·11	285	140·56
46	7·78	98	36·67	148	64·44	198	92·22	290	143·33
48	8·89	100	37·78	150	65·56	200	93·33	300	148·89
50	10·00	102	38·89	152	66·67	202	94·44	350	176·67
52	11·11	104	40·00	154	67·78	204	95·56	400	204·44
54	12·22	106	41·11	156	68·89	206	96·67	450	232·22
56	13·33	108	42·22	158	70·00	208	97·78	500	260·00
58	14·44	110	43·33	160	71·11	210	98·89	600	315·56
60	15·56	112	44·44	162	72·22	212	100·00	700	371·11
62	16·67	114	45·56	164	73·33	214	101·11	800	426·67
64	17·78	116	46·67	166	74·44	216	102·22	900	482·22
66	18·89	118	47·78	168	75·56	218	103·33	1000	537·78
68	20·00								

TABLE II

Comparison of Fahrenheit and Centigrade Thermometric Scales

Fig. 1 illustrates various forms of thermometer in common use.

The reading of an ordinary mercury thermometer is said to give the temperature of a gas, liquid or solid in which it is immersed, but in reality what it gives is something considerably more complicated when applied to air in buildings. It gives a temperature at which the heat received from the surrounding objects exactly equals the heat given off to other surrounding objects when a perfect balance is obtained, and this heat transference may be partly by conduction, partly by convection and partly by radiation (defined later).

If a mercury thermometer is dipped in a hot liquid, conduction and convection account for practically the whole effect, but when a thermometer is used for giving the temperature of a gas such as air, this is by no means true, as the effect of radiation may be quite important. Thus the shade temperature in the summer-time may be 80° F., while the sun temperature is 100° F. This does not mean that the temperature of the air is any different in the two cases, but in the former case the thermometer is not exposed to the radiation of the sun, and in the latter case it is.

This is of more than theoretical interest, for some forms of heating depend largely on radiation, and it then becomes rather important to have a

clear idea of what one is measuring when one talks about the temperature of the air. Is the effect of radiation to be included or excluded in measuring the temperature of the air? Suppose, for example, one is considering panel heating, taking the form of radiant heat from a warm ceiling. Is the tem-

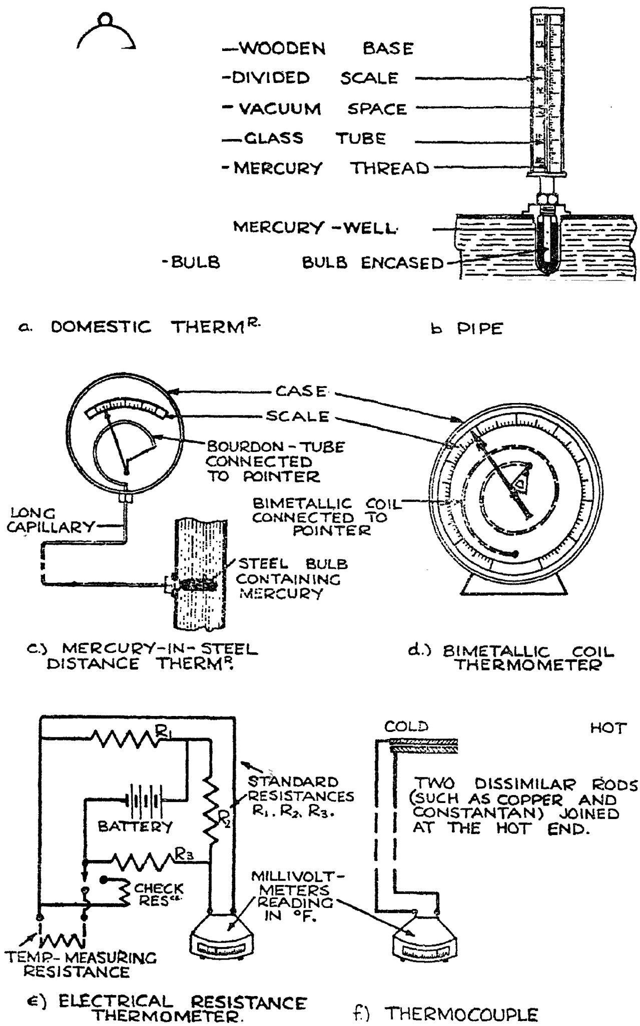


Fig. 1.—Types of Thermometer.

perature to be measured in such a way that the thermometer is screened from the source of radiant heat (in other words, the shade temperature), or is it to be measured by a thermometer which is exposed to the radiant heat (equivalent to the sun temperature)?

It is a matter of considerable interest, and very much simplifies our problem, that an ordinary mercury thermometer gives approximately the same reading in a room which has been allowed to reach stable conditions as regards temperature, whether it is screened from the source of radiant heat or not, provided that the source of radiant heat is one of relatively low temperature, such as is obtained with ordinary panel heating obtained by the circulation of warm water in pipes or large flat surfaces. This is because glass is practically impervious to the radiation from low-temperature sources.

This, however, is not at all true where high-temperature radiation, such as that received from the sun and from white-hot gas fires or electric radiators is concerned. This is really the explanation of the curious phenomenon of the ordinary glass greenhouse, which, as everyone knows, gets extremely hot inside when the sun is pouring down on it, and retains its heat long after the sun has ceased to shine upon it.

The air outside the greenhouse in the sun receives exactly the same amount of radiation from the sun as the air inside the greenhouse. Why, therefore, does that outside remain, say, at 70° when the air inside may be at 100°? In both cases the temperature of any body subject to the radiation goes up until the total quantity of heat received from the sun and all other surrounding objects is balanced by the heat lost to the other surrounding objects, and at first there appears to be no reason why the temperature reached in the two cases should be in any way different. The explanation of this is to be found in the peculiar property of glass in being pervious to high-temperature radiation and impervious to low-temperature radiation (a phenomenon sometimes referred to as *diathermancy*). The heat from the sun (which is a source of high-temperature radiation) passes through the glass and warms the objects therein contained, which are unable to radiate this heat back through the glass.

It will be seen that the exact meaning of temperature requires a good deal of consideration when problems of heating or air-conditioning are under discussion.

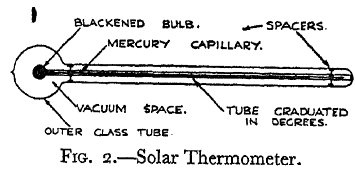


FIG. 2.—Solar Thermometer.

A thermometer known as a *solar thermometer* is employed when it is desired to obtain a reading which includes the effect of solar radiation (see Fig. 2). This thermometer consists of a glass bulb containing a vacuum, in the middle of which is an ordinary thermometer bulb with a blackened surface. The idea is that the thermometer bulb will not be affected by the air temperature, from which it is insulated by the intervening vacuum, but will receive only radiation, and so will measure the radiant heat only. This gives quite satisfactory readings when applied to the measurement of sun temperatures.

In the early days of panel heating attempts were made to use such a

thermometer to measure the radiant heat from the panels inside buildings, and practically identical results were obtained with this thermometer and with an ordinary mercury thermometer. The explanation is again to be found in the fact that though the glass envelope permits high-temperature radiation to pass, it is almost impervious to low-temperature radiation. The outer bulb takes up nearly the same temperature as the bulb of an ordinary mercury thermometer, i.e. the temperature of the air, and this temperature is gradually transferred into the central bulb by radiation from the inner surface of the enclosing envelope. The bulb is screened from the low-temperature radiation just as much as the mercury in an ordinary thermometer is screened from it.

Dr. H. M. Vernon, in his book *The Principles of Heating and Ventilation*, describes a copper globe thermometer of his invention consisting of a 6 in. hollow ball of copper blackened on the outside, with an ordinary thermometer projecting into the middle of it (see Fig. 3). This receives radiation from low-temperature sources and gives readings higher than the air temperature recorded by an ordinary mercury thermometer when in the presence of high- or low-temperature radiating surfaces. With this instrument he obtained temperatures 12·8° F. above those recorded on an ordinary thermometer when 5·8 ft. from a gas fire, which difference came down to 4·6° at a distance of 9·1 ft. when the angle subtended by the fire was much less.

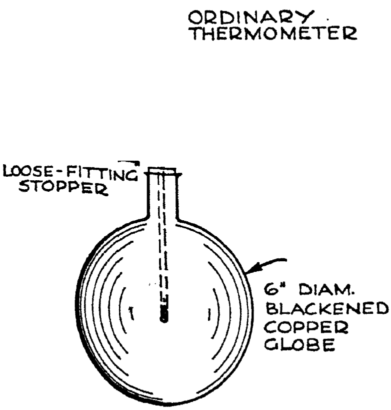


FIG. 3.—Globe Thermometer.

In the case of ceiling panels the corresponding effect was only 1·6° at a distance of 9 ft. from the panel, and 1·2° 15 ft. away. It should, however, be noted that in a room of, say, 10 ft. clear height, the occupant's head may be as little as 5 ft. from the ceiling panel, and in that case the additional effect of radiation may be more than 2°.

In Chapter II, on the general principles of heating, mention will be made of the *Eupatheoscope*, which gives a measure of heat lost by an individual sitting in a room, taking into account the effects of radiation, air temperature and air movement. This measures something intimately connected with the comfort of an occupant, not of temperature alone.

Humidity—The humidity of the air may be defined as the quantity of water vapour which it contains, and the *absolute humidity* may be expressed

in terms of grains of moisture per cubic foot. The water vapour is not dissolved in the air, the two merely forming a physical mixture.

The quantity of water vapour which can exist in a given volume depends on the temperature, being far greater at high than at low temperatures, but varying only slightly with changes of pressure over the range normally encountered in atmospheric conditions. At low pressure the volume of vapour required for saturation is slightly less than at high pressures.

Table III shows how the number of grains which a cubic foot of air will contain when saturated, varies with the temperature.

TABLE III

GIVING HUMIDITY OF SATURATED AIR, IN GRAINS OF WATER-VAPOUR PER CU. FT. OF AIR AT VARIOUS TEMPERATURES

Temperature ° F.	Grains.	Temperature ° F.	Grains.	Temperature ° F.	Grains.
30	1·94	54	4·73	78	10·37
32	2·12	56	5·05	80	11·03
34	2·28	58	5·42	84	12·47
36	2·46	60	5·80	88	14·04
38	2·66	62	6·20	92	15·81
40	2·86	64	6·62	96	17·77
42	3·08	66	7·07	100	19·91
44	3·31	68	7·54	104	22·30
46	3·56	70	8·06	108	24·89
48	3·82	72	8·58	112	27·73
50	4·09	74	9·16	116	30·90
52	4·40	76	9·74	120	34·31

Values independent of pressure. Based on *I.H.V.E. Guide*.

If air at a high temperature is more or less saturated with moisture and the temperature is then lowered, it reaches a point where it is unable, at the lower temperature, to contain the same number of grains per cubic foot as before, and the excess moisture becomes visible in the form of mist, dew or rain. The ratio * of the quantity of moisture actually contained in the air at a given temperature, to the quantity which would saturate it at that temperature (see Table III) is known as the *relative humidity*. A relative humidity of, for example, 50 per cent. therefore tells us nothing whatever as to the number of grains of moisture per cubic foot, but only that the air at any particular temperature contains half as much moisture as would have saturated it at that temperature.

The relative humidity is, however, the condition of air upon which depends, more than anything else, the rate at which evaporation from a moist surface will occur. Thus at 100 per cent. relative humidity no

* The ratio is strictly that of the two vapour pressures, but the differences are slight at normal temperatures.

evaporation will occur, while at 50 per cent. relative humidity the evaporation will be rapid. The human body provides such a moist surface, and the evaporation which takes place from it with low relative humidities produces various physiological effects, such as a parchiness in the throat and a cooling effect on the face and hands, especially in the presence of air movements, thus producing discomfort.

Too high a relative humidity produces other signs of discomfort, such as lassitude, caused primarily by the inability of the skin to rid itself of moisture and heat.

The normal evaporation from the skin is accompanied by a cooling effect on the body owing to the great heat required to convert water into water vapour (equivalent to approximately 970 B.T.U.'s per lb.) (see also *latent heat*, page 16), and it is on this that the cooling effect of a current of air principally depends. With excessive humidity, when this evaporation does not occur, the cooling effect clearly disappears also, and discomfort arises at high temperatures.

This explains what travellers in tropical climates have experienced for many years, that a high air temperature with a low relative humidity may be borne more easily than a lower air temperature in very humid conditions.

In cold climates the opposite is, however, the case. Air at low temperatures has a greater feeling of coldness at high relative humidities than at low. The reason for this effect on the sensation of cold in cold weather and of heat in hot weather, being both greatly accentuated in a humid atmosphere (within the everyday experience of our cousins from Canada as regards cold, and those from the tropics as regards heat), is due to the changes which high humidity induce on the human skin. In a dry atmosphere, the skin dries up and hardens and becomes more insulating, and so feels the cold or heat less.

Conversely, in a moist atmosphere the skin swells, the pores open, the skin becomes more conducting and sensitive. This is an important effect, and the Eupatheoscope or any similar instrument can not take it into account.

It therefore appears that high relative humidity causes discomfort both at high and low temperatures, in the former case by producing a sensation of extreme heat, and in the latter case of extreme cold. The temperature at which no change in sensation occurs with change of humidity is stated to be 46° F. for still air and 51° to 56° F. for air moving at speeds of 100 to 500 ft. per minute respectively.

Those who have spent the winter in Swiss winter sports hotels will have had experience of the effect produced in the evenings, when the temperature outside is about 32° below freezing and the temperature inside about 70° F. Any air entering the rooms comes in perhaps saturated with moisture (see Fig. 4), but saturated at so low a temperature that its absolute humidity is extremely low, and when its temperature is then raised from 0° to +70° its relative humidity becomes exceedingly low also [5 to 6 per

cent. R.H.]. This produces conditions which can fortunately be partially relieved by the consumption of liquid refreshment.

Humidification and De-humidifying—From what has been said it will be clear that in very hot summer weather the hot air from outside coming into a cool building will have its relative humidity raised, and will therefore produce oppressive conditions. Sometimes the relative humidity may be raised to 100 per cent., and condensation will occur.

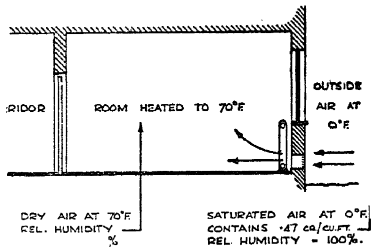


Fig. 4.—Showing Dryness of Air in Winter due to Heating.

Comfortable conditions may therefore be produced by some device whereby the temperature and relative humidity may be reduced, and this is known as de-humidifying. Most de-humidifying processes depend on allowing the air to come in contact with very cold surfaces where the excess moisture is deposited, and then warming it again to a comfortable temperature before distribution.

In cold winter weather the opposite occurs. Cold air from the outside comes into a warm room, and is warmed by the heating system to a higher temperature accompanied by a drop in the relative humidity. In such cases there is need for humidification, i.e. for adding to the moisture content of the air. This is generally effected by passing the air through a washer in which it is intimately mixed with water in the form of a very fine spray or mist.

In rough figures, it may be said that to most people a comfortable humidity is between 50 and 70 per cent. combined with a temperature of somewhere between 62° and 66° F.

Conduction, Convection and Radiation—It will be important, in the discussions that follow, to understand clearly the difference between the three main methods of transferring heat from one body to another.

Conduction may be described as a heat transfer from one particle to another by contact. If, for example, a hot lump of copper is placed in contact with a cold lump of copper, as in Fig. 5, heat is said to be conducted from the hot to the cold, until the two will finish at an intermediate temperature which may be calculated by equating the total weight times the final temperature with the weight of the hot body times its temperature added to the weight of the cooler body times its temperature. If the two bodies are of different material, then the weight of each has to be multiplied by its specific heat before this calculation

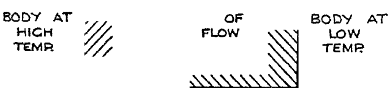


Fig. 5.—Conduction.

TABLE IV
Properties of Materials

Material.	Specific Gravity. (Water = 1·00.)	Specific Heat. (Water = 1·00.)	Coeff. of Linear Expansion per 1° F. $10^{-6}\times$...	Conductivity: B.T.U.'s/1° F/ Sq. Ft./Hr./ 1″ Thickness.
Copper (Sheet)	8·8	0·093	9·7	2670
Aluminium (Cast)	2·71	0·24	13·0	1460
Magnesium	1·74	0·24	14·2	1060
Zinc (Cast)	6·86	0·095	16·3	950
Brass (Cast)	8·1	0·095	10·5	630
Cast Iron	7·2	0·125	6·2	480
Mild Steel	7·85	0·117	6·3	325
Tin	7·3	0·056	15·0	430
Lead	11·4	0·031	15·8	230
Mercury	13·6	0·033	33·3	45
Granite	2·64	0·215	4·4	20·3*
Marble	2·72	0·215	6·1	17·4*
Concrete (1:2:4 mix)	2·4	0·20	5·5	7·0*
Limestone	2·18	0·205	3·5	10·6*
Slate	2·56	0·18	10·8	10·4*
Asphalt	2·24	—	—	8·7*
Brickwork (Fletton)	1·8	0·19	1·2	8·0*
„ (Typical)	—	—	—	5·0†
Firebrick	2·0	0·20	2·7	7·4 (at 800° F.)*
Glass (Sheet)	2·50	0·20	4·7	7·3*
Tiles (Burnt Clay)	1·93	0·20	—	5·8*
Plaster	1·4	0·20	—	4·0*
Asbestos Millboard	1·02	0·195	—	1·92*
Asbestos Cement	1·54	0·20	5·5	1·90*
Oak	0·77	0·57	Mean 2·2	1·11*
Pitch-pine	0·66	0·53	Mean 2·2	0·96*
Deal	0·59	0·46	Mean 2·2	0·87*
Plaster Board	0·97	0·20	—	1·1*
Diatomaceous Bricks	0·48	0·19	0·8	0·60*
Cellular Concrete	0·40	0·20	0·8	0·45
Magnesia, 85%	0·19	—	—	0·40*
Fibre Board (Tentest)	0·28	—	—	0·38*
Cork Board	0·14	0·43	—	0·30
Hairfelt	0·17	—	—	0·30*
Glass Fibre	0·14	0·195	—	0·28*
Water 39° F.	1·000	1·004	0·0	4·0
Water 60° F.	0·999	1·000	36·0	4·0
Water 212° F.	0·955	1·005	138·0	4·0
Ice	0·92	0·504	29·0	16·0
Air (at Normal Pressure) 32° F.	0·0013	Const. Press. ·240 Const. Vol. ·168	677·	0·18
Air (at Normal Pressure) 212° F.	0·00095	Const. Press. ·240 Const. Vol. ·168	495·	0·18

The above figures are average values at temperatures of normal use.

* From *I.H.V.E. Guide.*

† From *A.S.H.V.E. Guide.*

Other data taken from standard authorities.

Conductivity is the property of the bodies of being able to conduct heat readily, and the measure of conductivity is the number of heat units transmitted per degree difference per unit thickness and per unit face area, in unit time.

Table IV gives conductivities of various metals, building materials and insulators, and it will be seen that metals have a high conductivity, while materials like brick have a low conductivity. Certain bodies have such a low conductivity that they are known as *insulators*, and are used for covering warm surfaces to prevent heat losses from them. For convenience Table IV also gives the specific gravity, specific heat and coefficient of expansion of these substances. These terms will be referred to later.

Good conductors of heat are generally good conductors of electricity, though there is no rigid relationship between the two.

The conductivity of many materials varies considerably with temperature, and therefore figures should only be used within the range to which they apply. The insulation value is inversely proportional to the conductivity.

Porous materials are bad conductors when dry and good conductors when wet, a fact which is sometimes lost sight of when attempting to warm a newly-constructed building, where the heat losses may be far higher in the early months than they will be when it has properly dried out.

Convection is a transfer of heat which involves the movement of hot particles of a fluid medium from a hot body to a cold. A common illustration may be found in the ordinary radiator (so-called) (see Fig. 6*a*). This warms the air immediately in contact with it, which expands and so becomes lighter than the rest of the air in the room. It consequently rises, forming an upward current from the radiator, and travels to the colder portions of the room, to which it gives up its heat and eventually returns to the radiator for the process to be repeated.

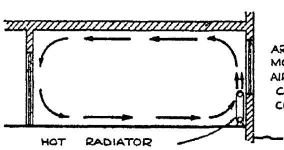


FIG. 6*a*.—Convection.

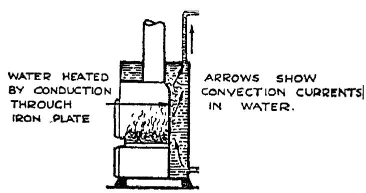


FIG. 6*b*.—Convection in Water.

Another example is the heating of water in a boiler, as shown in Fig. 6*b*. The water in contact with the hot surfaces over the fire becomes heated, expands, and produces an upward circulation, exactly as in the case of the air, eventually returning to the boiler for reheating. Convection, therefore, implies a medium capable of movement from the hot body to the cool body to be heated, and cannot occur in vacuum when no such medium exists.

Radiation is a phenomenon with which we are most familiar in its application to the problem of light.

In Newton's time the phenomenon of radiation was explained as a bombardment of infinitesimal particles from the source of heat, which were supposed to impinge on the cool body to be heated. At a later date radiation, whether of light or of heat, was supposed to be a wave action in an imaginary medium known as the ether, which was invented by mathematicians to account for this phenomenon. At the present time, and in the light of the latest discoveries, the whole theory of the ether has been demolished.

For our present purposes it is probably accurate enough to look on radiation as a transference of energy which takes place in rays in such a way that the intensity varies inversely with the square of the distance, and is independent of any substantial medium such as air, i.e. it occurs just as readily across a vacuum as across a room filled with air, and does not depend on warming the medium through which it travels.

Fig. 7 shows the order in the radiation scale in which radiant heat occurs.

The so-called 'radiator' transfers its heat partly by convection and partly by radiation, the proportion depending mainly on the shape of the

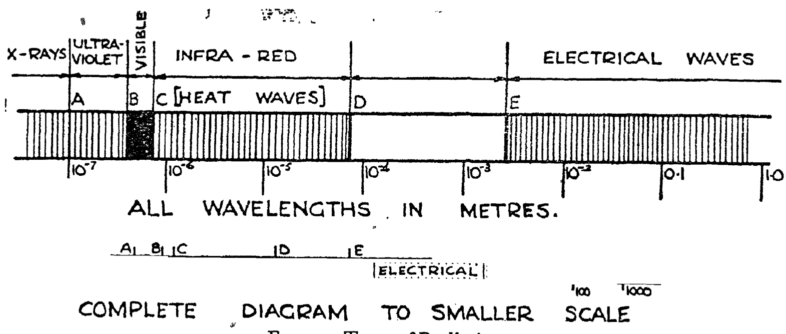


FIG. 7.—Types of Radiation.

radiator. Those surfaces which face one another radiate very little to the rest of the room and depend principally on convection. Also, if anything is done to obstruct the free flow of air over the radiator its proportion of convection is reduced. Flat surfaces fixed to the wall, however, such as Rayrads, may have a relatively high proportion of radiation to convection.

In very rough figures, the proportion of radiation to the total heat emission in ordinary radiators may be about 20 per cent., in a wall Rayrad about 50 per cent., and in a heating panel of the ordinary warm-water type on a ceiling about 90 per cent. This shows that the term 'radiator' is a misnomer, since approximately 80 per cent. of its heat is transmitted by convection.

Fig. 8 shows the relatively concentrated radiant beam emitted from an electric bowl fire and the diffused radiation from a warmed ceiling panel.

The amount of radiation emitted from a surface depends on its texture and colour. Dead black is the best radiator and polished metal the worst. This effect is dealt with more fully later. (See p. 179.)

Surfaces which are good radiators of heat are found to be the best absorbers also: thus a black-felted roof is often found to be covered with hoar frost on a cold night, due to its good radiation into space, whereas

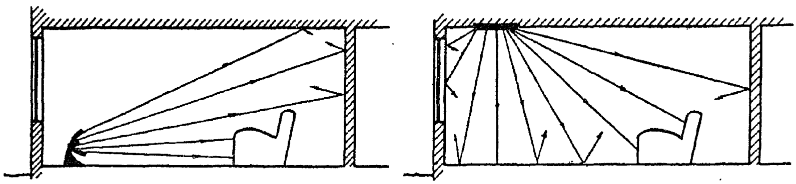


Fig. 8.

(*a*) Visible heat radiation from electric bowl-fire.

(*b*) Invisible radiation from heated ceiling panel.

other surrounding objects may be apparently unaffected. Similarly, a black suit of clothes on a warm day is found to absorb much more heat from the sun than one of white material such as is worn in tropical countries. Again, asphalt on a flat roof does not become nearly so hot in summer if painted white.

Specific Heat and the B.T.U. and Calorie defined—The *specific heat* of a substance is the quantity of heat required to raise unit weight of that substance through unit difference of temperature, the specific heat of water being 1·0. See Table IV (p. 10).

The *British Thermal Unit* (*B.T.U.*) is the quantity of heat required to raise 1 lb. of water through 1° F. In the metric system the unit of heat is the *calorie* and is equivalent to the raising of 1 gramme of water through 1° C., the *large Calorie* being for 1 kilogram raised through 1° C.

For a very strict definition of specific heat it is necessary to state at what temperature it is to be measured, because it sometimes happens that the quantity of heat required to raise a given weight of material is not exactly the same, say from 0° to 1° as it is from 100° to 101°; but while these differences may be important in a scientific work of the greatest accuracy, they will not trouble us in the ordinary problems connected with commercial heating and ventilating.

From Table IV it will be seen that next to water come timbers and similar materials, with a specific heat round about ·5. Most building materials, such as concrete, brick, stone, etc., come next with a figure of about ·2, whereas the various metals have figures round about ·1.

The only material we know as having a specific heat higher than water is liquid hydrogen, and this is only very slightly higher.

Expansion—It will be found that Table IV gives a column for the *Coefficient of Linear Expansion* of various materials.

All materials, with very few exceptions, expand on being warmed, and contract on being cooled. The Forth Bridge expands approximately 2 ft. as between warm and cold weather.

It is found that for most materials the expansion varies directly with the length and with the difference of temperature.

The coefficient of expansion for a material is defined to be the proportion of its original length which it lengthens with 1° F. rise of temperature. Thus the coefficient of expansion of mild steel is approximately ·000006, which means that if a bar is 1″ long, say, at 60° it will be 1·000006″ long at 61°. Similarly a pipe 1000″ long raised through 100° F. will be 1000·6″ long at the higher temperature. In other words, it has increased in length by ·6″, and so on in proportion.

It will be seen from this how necessary it is to make due allowance for the expansion and contraction of such things as long straight pipes, where the movement may be quite considerable and in practice pipes beyond a certain length of straight have to be provided with expansion joints, which will be dealt with in the appropriate place later. 'Invar' steel, a material much used in instruments, has no coefficient of expansion for a wide range of temperature.

The coefficients of expansion given in Table IV are average figures within the range of normal temperatures—that is, between 60° and 212°—but a word of warning is necessary in applying these figures, as in some cases they vary considerably beyond this range. Water is an example of this, and near the freezing point it has a negative coefficient of expansion, i.e. it actually expands on lowering the temperature from about 39° F. down to 32° F., the coefficient at 39° F. being approximately zero.

Superficial Expansion is the increase in area due to increase of temperature. The Coefficient of Superficial Expansion is taken as twice the linear coefficient. For an expansion of 'α' on the sides of a square of 1 unit initial length, increase in area $= (1+\alpha)^2 - 1 = 2\alpha + \alpha^2$. α^2 is negligible, hence coef. sup. exp. $= 2\alpha$. *Cubic Expansion* is the increase in volume due to increase of temperature. The Coefficient of Cubic Expansion is taken as three times the linear coefficient (by a similar argument to the above).

Specific Heat and Expansion of Gases—The perfect gas conforms to Boyle's and Charles' Laws, which state that the volume varies inversely with the pressure when the temperature is constant, and the volume varies directly with the absolute temperature when the pressure is constant. In other words, $\frac{PV}{T}$ is a constant where P is the pressure, V is the volume and T is the absolute temperature.

Now the number of degrees Fahrenheit between freezing point and absolute zero is 493 (see Table I). It therefore follows that if a certain volume of perfect gas is contained in an envelope and its temperature is

raised through 1° from 32° to 33° F. without altering the pressure, the increase in volume will be $\frac{1}{493}$ of its original volume. This gives a coefficient of cubic expansion of $\frac{1}{493}$, which is approximately 2030×10^{-6}.

If, however, a volume of gas at the boiling point of water is raised through 1°, then its absolute temperature being $212 + 461 = 673°$ F., its coefficient of expansion is $\frac{1}{673} = 1486 \times 10^{-6}$.

Most gases conform very closely to the properties of the perfect gas when at a temperature remote from their temperature of liquefaction. Ordinary air conforms very closely indeed, but CO_2, which can be liquefied at only moderate temperature reductions, does not conform quite so well.

Gases have two specific heats according as to whether the pressure or the volume is kept constant while the temperature is raised. If the pressure is kept constant then the increase in temperature is accompanied by an increase in volume, and the specific heat at 32° F. is ·238. If, on the other hand, the volume is kept constant, then the increase in temperature is accompanied by an increase of pressure, and the specific heat is ·168 at this temperature.

Mechanical Equivalent of Heat—Joule was the first to show, in the year 1840, that heat and mechanical energy are mutually convertible either way. He made experiments in which a measurable quantity of work was performed inside a closed and lagged vessel, and by observing the temperature rise of a known weight of material of known specific heat, he was able to discover how much heat corresponded to a given amount of mechanical energy.

The results of these experiments and many which have been conducted since show that one B.T.U. is equivalent to 778 ft. lbs., i.e. if the energy released by 1 lb. falling through 778 ft. against a resistance is collected and converted into heat, it will raise the temperature of 1 lb. of water by 1° F. Or, conversely, if the whole of the heat which is released by 1 lb. of water in dropping in temperature through 1° F. can be converted into mechanical energy in an engine of 100 per cent. efficiency, it will raise a weight of 1 lb. through 778 ft., or 778 lb. through 1 ft.

Horse Power—Power is the rate of expending energy and may be expressed in ft. lbs. per minute.

A horse power was supposed to be the average power of the horse in dragging loads, and was defined as being the rate of expending energy corresponding to 33,000 ft. lbs. per minute, or 550 ft. lbs. per second. Actually this measure is quite an arbitrary one and does not correspond well with what a horse can do. This, however, is past history, and the horse power in these matters has this clearly defined meaning. It will therefore be seen that a horse power may be expressed in terms of heat, and is equivalent to 42·4 British Thermal Units per minute, or 2544 B.T.U.'s per hour.

It is found that an ordinary human being, by maintaining an internal temperature of approximately 98·4° F., emits approximately 400 B.T.U.'s of heat per hour when at rest at normal temperatures. This heat

goes out partly as radiation and convection, partly as warm air containing water vapour from the lungs, and partly as evaporation from the skin. It will be seen from this that a human being is developing nearly $\frac{1}{6}$ h.p. merely by being alive. This is equivalent to the no-load loss of an engine, i.e. the amount of heat expended when the engine is performing no external work.

When a man is doing physical work at the rate of 66,000 ft. lbs. per hour, which is one-thirtieth of a horse power, it has been shown that the amount of heat expended by him goes up to approximately 1300 B.T.U.'s per hour. Now one-thirtieth of a horse power is equivalent to approximately 85 B.T.U.'s per hour, and it is therefore equivalent to saying that to get 85 B.T.U.'s per hour of useful work out of a man he has to expend 1300 B.T.U.'s in wasted heat, so that his efficiency is $\frac{85 \times 100}{1385}$, which is approximately 6 per cent. This is lower than that of the worst engine in common use, and should conduce to humility. A man can exert still more energy for short periods, and will then emit correspondingly greater quantities of heat.

Electrical Energy—There is an electrical equivalent of heat, just as there is a mechanical equivalent; that is to say, heat energy, mechanical energy and electrical energy are all mutually convertible.

The unit of electrical energy is known as the *Watt*, and represents the power exerted in a circuit when a quantity of electricity known as one *ampere* flows through it at a pressure or potential difference of one *volt*. Watts equals amperes × volts.

A thousand of these small units are known as the *kilowatt*; 746 watts are equivalent to one horse power. Owing, however, to the losses in actual motors, more than 746 watts have to be put into a motor to get 1 horse power of mechanical energy out of it, depending on the efficiency. In rough figures it is generally safe to take 1 kilowatt as necessary to produce 1 horse power for small motors, though for larger and more efficient machines the figure may lie anywhere between 1000 and 800 watts.

A **Board of Trade Unit** (generally known simply as *a unit*) of electricity is 1 kilowatt maintained for 1 hour, and may represent 10 amperes flowing in a circuit at 100 volts pressure for one hour, or 100 amperes flowing in a 10 volt circuit for an hour, and so on.

The heat equivalent of 1 watt for one hour is 3·415 British Thermal Units, and of 1 kilowatt for one hour is 3415 B.T.U.'s. Thus 1 unit of electricity is equivalent to the dissipation of 3415 B.T.U.'s. In other words, if 1 unit of electricity is entirely expended through a resistance in warming water, it will warm 3415 lbs. of water through 1° F.

Latent Heat—If water is heated from freezing point to boiling point (i.e. through 180° F.), 180 B.T.U. have to be supplied to it per pound, by definition. If the supply of heat be continued it is found that the temperature does not increase (assuming the vessel is open to atmosphere), but the water is gradually converted into steam at the same temperature. The experiment

will show quite clearly that the amount of heat absorbed to effect this conversion is extremely large in comparison with the heat required to warm the water, and in fact it requires 970·6 B.T.U.'s to convert 1 lb. of water at boiling point into 1 lb. of steam at the same temperature.

It will be seen that this is approximately 5·37 times as great as the heat required to warm the water from freezing point to boiling, and this gives some idea of the importance of this phenomenon. This large quantity of heat required to convert a unit weight of any substance from the liquid to the vapour state is known as the *latent heat of vaporization.*

The reverse is equally true; when a pound of steam at 212° condenses so as to produce 1 lb. of water at the same temperature 970·6 B.T.U.'s are liberated. Of all known substances, water has the greatest latent heat.

The boiling point of water varies with the pressure; at high pressures the boiling point is greatly raised and, conversely, it is reduced at pressures below atmospheric. It is often convenient to add the heat required to raise the water from freezing point to the temperature at which it is vaporized to the latent heat of vaporization, and this quantity is known as the *total heat of steam* at that temperature. Thus the total heat of steam at 212° is 970·6 plus 180 = 1150·6 B.T.U.'s. A table of total heat of steam is given for various pressures in Chapter XIV under the heading 'Properties of Saturated Steam', and is necessary for many calculations. It will be found from this that the latent heat becomes less as the pressure increases, whilst at the same time the total heat increases due to the higher temperature.

There is also a *latent heat of solidification* when water is converted into ice. The reverse phenomenon brings into play the *latent heat of fusion* as compared with latent heat of evaporation, and for water has the figure of 144 B.T.U.'s. Here again it will be noticed that it takes quite a large quantity of heat to melt 1 lb. of ice *without altering its temperature.*

In all problems of humidification and de-humidifying where water is either evaporated or condensed, the latent heat is the principal factor concerned.

All materials have a latent heat of fusion and one of vaporization, but these have lower values than in the case of water, and, of course, the temperatures at which the changes of state occur may be widely different.

Availabity of Heat—From the definition of the B.T.U. it is clear that 1000 lbs. of water at a temperature of 1° F. above the surroundings contain 1000 B.T.U.'s, as also does 1 lb. of water at a temperature of 1000° above the surroundings (assuming for the moment that it is in a vessel that enables it to be at that temperature), but while the Heat Content in B.T.U.'s is the same in both cases, the *value* of this 1000 B.T.U.'s is very different. In the former case it is generally quite impossible to make any effective use of the heat, while in the latter case nearly the whole of it could be converted into mechanical energy or used for heating or hot-water supply. From this it appears that the value of heat depends not only on the number of B.T.U.'s contained, but also on the temperature above

the surroundings. Indeed, a new unit is really required which ought to be given some suitable name. It should be the product of the number of B.T.U.'s and the temperature difference above normal. If such a unit were in general use, many misconceptions would immediately be swept away, which are due to ascribing the same value to B.T.U.'s at an unusable low temperature and the same number of B.T.U.'s at a useful high temperature. It is similar to the conception of entropy, but in this case in reverse.

Table V gives some useful physical constants.

TABLE V

VARIOUS PHYSICAL CONSTANTS AND EQUIVALENTS

1 pound (avoirdupois)	=16 oz.	=256 drachms	=7000 grains
		=1·21 lb. troy	=·4536 kg.
ton ,,	=2240 lbs.	=1·12 tons (U.S.)	=1016 kg.
kilogramme	=1000 gram.	=15,430 grains	=2·205 lbs.
cubic foot	=6·236 galls.	=62·36 lbs. water at 62° F.	
gallon (Imperial)	=4 quarts	=8 pints	=4·55 litres
	=277 cub. in.	=10 lbs. water at 62° F.	
pint ,,	=20 fl. oz.	=0·57 litres	=34·7 cub. in.
gallon (U.S.)	=231 cub. in.	=0·84 Imperial galls.	
	=8·4 lbs. water at 62° F.		
litre	=61 cub. in.	=0·22 Imperial galls.	
B.T.U.	=1 lb. water raised through 1° F.		
	=778 ft. lbs.	=·252 Cals.	
Calorie (large)	=1 kg. water raised through 1° C. =3·97 B.T.U.		
B.T.U. per sq. ft.	=2·71 Calories per sq. metre		
horse power	=0·746 kw.	=33,000 ft. lbs./min.	=2544 B.T.U./hour
watt	=1 joule per sec.	$=10^7$ ergs per sec.	
kilowatt-hour	=1 unit	=1000 watts for 1 hour	
	=3415 B.T.U.	=2,654,000 ft. lbs.	
1 lb. per sq. inch	=703 kg./sq. metre	=2·04″ mercury	
	=2·32 ft. or	=27·8″ water at 62°	=1887 ft. air at 62°
1 ft. water at 62°	=0·432 lbs./sq. in.	=0·88″ mercury	
		=816 ft. air at 62°	
Latent heat of water at atm. pr.:	Evapn.	=970·6 B.T.U./lb.	
	Fusion	=144 ,,	

CHAPTER II

General Problems in Connection with Heating

1. CALCULATION OF B.T.U.'s REQUIRED TO MAINTAIN A GIVEN TEMPERATURE IN A BUILDING

In order to design a heating system for any given building, it is first necessary to ascertain the amount of useful heat to be supplied to each room to provide an equable temperature. Heat is required (i) to raise the temperature initially and (ii) to maintain this temperature under steady conditions against the constant loss of heat which occurs to the outside air. The second of these, (ii), is the basis on which the amount of heat to be supplied is arrived at by calculation. Item (i) is ignored in this computation, as the warming up may in most cases extend over a long period. This brings in the question of time lag, which is referred to later (see p. 51).

Heat is lost from a warm room to cooler surroundings in two ways:

(*a*) by transmission through the building materials to outside or to unwarmed rooms;

(*b*) by actual escape of warmed air.

The losses due to the former, (*a*), are often termed *radiation losses*, though they are more correctly described as conduction or *transmission losses*, since they involve the transference of heat through solid matter by conduction. These losses are also sometimes described as *transmittance* losses.

Those due to the latter, (*b*), are termed *infiltration losses*, because the escaping warmed air is replaced by cool air from outside, and this in turn requires heating to room temperature.

The sum of (*a*) and (*b*) are known as *heat losses*.

(*a*) **Transmission Losses**—The heat escaping by transmission may be calculated by multiplying the area of surface (wall, window, roof, floor, etc.) in sq. ft., by the coefficient of transmission in B.T.U.'s per sq. ft. per deg. difference of temperature per hour (U).

The result requires to be multiplied by the appropriate temperature difference between inside and outside, the loss being proportional to the temperature difference.

The area of each type of exposed surface, i.e. window, wall, roof, floor, etc., is either measured from drawings or from the building.

The Coefficient of Transmission, U, for the appropriate material and thickness may be taken from Table VI, which gives a list of these factors for various types of construction in common use.

TABLE VI

TRANSMISSION COEFFICIENTS. '*U*'

WALLS

	Thickness of Brick				
		9″		8″	22½″
Solid and Cavity Constructions.					
*Fletton brickwork, unplastered	·64	·47	·37	—	—
,, ,, plastered	·57	·43	·35	·29	·25
†Typical brickwork, unplastered	·59	·39	·29	·23	·20
plastered	·53	·36	·27	·22	·19
plus 4″ stone facing plastered inside	·44	·32	·25	·20	·17
,, 8″ ,, ,, ,,	·38	·28	·22	·19	·16
lined ¾″ wood on battens	·29	·23	·19	·16	·14
,, 1″ cork and plastered	·19	·16	·14	·13	·12
,, ⅝″ fibre board on battens	·25	·20	·17		
lined 'Ardor' (al. foil) and wallboard	·18	·15	·13		

	Overall Thickness excluding Finish			
		11″		20″
†Brickwork with 2″ unvented cavity plastered		·27	·22	
,, with 'Ardor' (al. foil) in cavity		·15	·13	·12
,, 4½″ with 2″ cavity and 3″ foamed slag concrete (light mix)	·20	—	—	

	Thickness of Concrete				
	4″	6″	8″	10″	12″
*Concrete, unplastered	·75	·61	·52	·45	·40
with ½″ plaster	·65	·55	·47	·42	·38
lined ¾″ wood on battens	·32	·29	·27	·25	·23
,, 1″ cork and plastered	·21	·20	·19	·18	·17
,, ⅝″ fibre board and battens	·26	·24	·23	·21	·20
,, 'Ardor' (al. foil) and wallboard	·20	·19	·18	·18	·17
backed 4½″ brick (used as shuttering) and plastered	·41	·37	·33	·30	·28
foamed slag concrete (light mix) plastered	·20	·19	·18	·16	—

	Thickness of Stone		
	12″	18″	24″
*Limestone (solid), plastered inside	·50	·40	·33
4″ hollow tile, rendered outside, plastered inside		·50	
,, ,, ,, ,, ⅝″ fibre board on battens inside		·23	
,, ,, 4½″ brickwork facing outside and plastered inside		·35	
4″ glass block wall (hollow)		·44	

Framed Constructions.	
¾″ weatherboards on studding lined lath and plaster or ¾″ wood	·44
Corrugated iron	
,, asbestos cement	1·4
,, protected metal	·9
Flat asbestos cement sheets	1·0
Corrugated sheeting (any type), lined ⅝″ fibre board	·31
¾″ wood	
plaster board or asbestos cement	·53
'Ardor' (al. foil) and plaster board	·13

† Based on *A.S.H.V.E. Conductivities*. See Table IV. Recommended for normal use for brickwork.

DOORS AND WINDOWS

*Solid door, $1\frac{1}{2}''$ thick timber	
,, ,, $1''$,, ,,	
*Glazed door, upper half glazed	
*Single window	1·0
*Double window	·50
Thermolux glass	·70

FLOORS

	Room Temperature 55° F.	60° F.	65° F.	70° F.
Floors in contact with Earth, Hardcore, etc. (assumed at 50°).				
Soil floor		·20	—	
Concrete, bare or with hard finish	·06	·13	·20*	·26
,, with screed and wood blocks or boards on fillets	·05	·10	·15*	·20
Floor with Ventilated Cavity Under.†				
Bare boards on joists, air-brick one side	·18	·24	·30*	·36
Ditto with parquet, lino or rubber	·15	·20	·25*	·30
Bare boards on joists, air-brick more than one side	·24	·32	·40*	·48
Ditto with parquet, lino or rubber	·21	·28	·35*	·42
Ice (as in skating rink)			1·30	

Intermediate Floors.	*Heat Flow Downwards*	*Heat Flow Upwards*
*Wood floor on joists, plaster ceiling	·22	·29
*$6''$ concrete, $2''$ screed	·43	·54
* ,, ,, with wood flooring	·30	·35
*Hollow tile floor, $6''$ thick,	·33	·40
* ,, ,, ,, with wood flooring	·25	·29

ROOFS AND CEILINGS

Flat Roofs.	*Thickness of Concrete* $3''$	$4''$	$5''$	$6''$	$8''$	
Reinforced concrete asphalt or roof sheeting outside, plaster inside	·63	·58	·53	·50	·44	
Ditto without plaster	·72	·66	·60	·57*	·48	
Ditto with $2''$ moler insulation (or $1''$ cork and plaster)	·21	·21	·20	·20		·18
Ditto ,, $2''$ cork insulation, and plaster	·13	·12	·12	·12	·11	·11
Ditto ,, $\frac{5}{8}''$ fibre board insulation	·33	·32	·31	·30	·29	·26
Ditto ,, $\frac{5}{8}''$ fibre board, without plaster	·35	·34	·33	·32	·30	·28
Ditto ,, 'Ardor' (al. foil), plaster on metal lathing with air gap				·18		
Reinforced concrete, asphalt, with suspended ceiling under	—	·35	·34	·33	·31	·29

$6''$ hollow tile or hollow concrete beams with asphalt or roof sheeting outside, plastered inside	·44
Arched rib construction with asphalt outside	·75
Ditto with sub-ceiling under	·35
Hollow or rib construction as above with $2''$ moler or $1''$ cork under asphalt	
Hollow or rib construction with $\frac{5}{8}''$ fibre board under asphalt or in lieu of plaster	

Pitched Roof.

*Corrugated iron	
asbestos cement	
protected metal	
sheeting (any type), lined $\frac{5}{8}''$ fibre board	
,, ,, ,, $\frac{1}{2}''$ wood	
,, ,, ,, plaster board or asbestos cement	·53
,, ,, ,, 'Ardor' (al. foil) and plaster board	

* *I.H.V.E. Guide.*

† For cavity floors with panel heating see note on p. 38.

TABLE VI. ROOFS AND CEILINGS (*Continued*)

*Tiles or slates, unlined, no ceiling	1·50
lined boards and felt or paper	·56
unlined with plaster ceiling	·56
lined boards and felted with plaster ceiling	·32
boards and felted with plaster ceiling and attic boards	·20
on battens, $\frac{5}{8}''$ fibre board under rafters, no ceiling	·41
on battens, $\frac{5}{8}''$ fibre board under rafters with fibre board ceiling	·20
boarded with plaster or plaster board ceiling with 'Ardor' (al. foil) insulation above	·13
Roof Glass.	
Single roof light	1·1
" " " 'Thermolux' glass	·75
Roof light and lay-light	·55
Pavement lights or 'Lenscrete' lights	·85

RETAINING WALLS

(*backed by earth at* 50°)

Brick or concrete up to 12″ thick	·07
" " over 12″ thick	·05

Alternatively the coefficient may be calculated from the formula:

where

U=heat transmission in B.T.U.'s per sq. ft. per degree difference between internal and external temperatures,
K_1=inside surface coefficient, B.T.U.'s per sq. ft.,
K_2=outside surface coefficient, B.T.U.'s per sq. ft.,
X_1, X_2, etc.=thickness of the various materials in inches,
C_1, C_2, etc.=conductivity of the various materials per sq. ft. per inch thickness, per hour, per degree F. difference.

Values of C for a variety of materials are given in Table IV.

For composite units such as hollow-tiles, the value C_t of the conductivity has to be found experimentally for each individual case, and is then expressed in B.T.U./hr./sq. ft./1° F. for the thickness of unit considered. In these cases the term $\frac{1}{C_t}$ would be used in place of the term $\frac{X_1}{C_1}$ in the above formula.

It is convenient in calculation to call the terms $\frac{1}{K}$, etc., and $\frac{X_1}{C_1}$, etc., the *Resistances* of the surfaces and component materials.

i.e. Overall Resistance $R=R_1+R_2+R_3+\ldots$

=the sum of the Resistances of the surfaces and component materials,

$$T=\frac{1}{R}.$$

The values of the surface resistances vary with the type of surface and degree of roughness, and with the air movement over the surface. For the

* *I.H.V.E. Guide.*

purpose of calculating values of U, the following figures may be taken for normal cases:

$$K_1 = 1{\cdot}65 \text{ B.T.U./sq. foot/hr.}$$

The values of U given in Table VI are subject to the following notes:

(*a*) Coefficients for brickwork quoted from the *I.H.V.E. Guide to Current Practice* are not felt by the Authors to represent the transmission from brickwork encountered in practice, and they recommend the use of the lower range of figures given in the table.

(*b*) The wall and roof coefficients published in the *I.H.V.E. Guide* are stated differently for various exposures. This is felt to be difficult to apply in practice, and where such coefficients are used in Table VI they have been taken for one condition only. The allowance for exposure is, in the Authors' opinion, more simply dealt with by the addition of a percentage to the total room heat losses as given under the heading 'Exposure' (see page 27).

It will be noted from the above formula that surface coefficients are included for the inside and outside surfaces. If the material is divided by an air gap, as in the case of a brick cavity wall, two further surfaces are introduced, thus reducing the value of U; the resistance of a normal 2″ air-gap may be taken as 0·91.

The meaning of surface coefficients may perhaps be better appreciated from Fig. 9.

This figure gives diagrammatically the curve of temperature change between the outside and the inside of a room separated by a pane of glass. It will be seen that the temperature gradient across the thickness of the glass is considerably less than the total difference of temperature between the outside and the inside, because there is a film of air in contact with the inside and outside faces which impedes heat losses. In strong winds this film is removed, and so the heat transmission is increased. Similarly in rainy weather heat is transmitted to a constantly replenished supply of cold water on the outside surface, and this also increases the transmission.

The coefficients for transmission through floors have been adjusted to permit of direct use in calculation of heat losses according to the room temperature. It is generally assumed that the earth under a building is at a temperature of 50° F. in winter in England. Thus in making a heat loss calculation for a room with a floor placed on the earth, this portion should, strictly, be taken out separately, and multiplied by room temperature − 50°. In order to save time, however, the same result is achieved by reducing the coefficient in proportion so that the result may be included in the total sum, which is all multiplied by the total temperature difference.

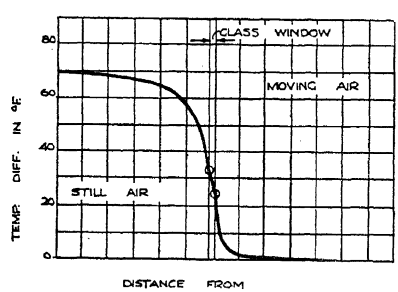


FIG. 9.

(*b*) **Heat required for Air Change**—In addition to the heat required to balance the radiation losses, it is obvious that the air in every ordinary room or enclosed space is subject to a gradual replenishment, partly by infiltration through porous walls, and partly by leakage past windows which are never completely airtight. There is in addition the necessary opening, either permanently or intermittently, of doors and other necessary means of access both for people and goods, supplemented, of course, by a certain amount of window being opened when the concentration of people in the rooms is at all appreciable.

The first of these, namely, the *infiltration through the walls*, obviously depends on the construction. The porosity of walls is much greater than is commonly realized.

Expressed in cubic feet per square foot per hour, it appears that a 9 in. wall of porous brick may allow infiltration as follows:

Infiltration through 9 in. brick wall in cu. ft. per hour per sq. ft.

Wind 5 miles per hour	1·75 cu. ft.
10 " "	4·2
15 " "	7·85
20 " "	12·2
25 " "	18·6
30 " "	22·9

If the wall is plastered the figures approximate to one-hundredth of the above, and for a 13 in. wall the differences are not important.

It will be seen that in high winds the infiltration is very considerable. If the inside surface is painted or varnished the infiltration is negligible.

There is no point in stressing the accuracy of these figures, as obviously they depend tremendously on the porosity of the brick, the nature of the mortar and other similar matters.

For windows, a method of estimation which has been used considerably in America, is to take the length of crack between window and window frame multiplied by an assumed thickness of crack of $\frac{1}{16}$ in. and various factors depending on the type of window.

In applying this method of estimation it is usual only to measure the foot run of crack on the windows facing in one direction, as a high wind will be definitely directional, and will only blow on one side, and the air which enters here has to find its way out from the corresponding cracks on the other wall face.

Expressing the results in cubic feet per lineal foot of crack per hour the average figures may be said to be roughly as follows:

Leakage of air in cu. ft. per hour per lineal foot of crack of window opening

5 miles per hour	20 cu. ft.
10	45
15	70
20	96
25	125
30	

The figures for well-fitting windows will obviously be less than this and for loosely-fitting windows correspondingly greater, but there seems little object in pursuing this matter in detail, because until windows are standardized no such table would be sufficiently applicable in any particular case to enable definite figures to be stated without making tests in each individual case. With timber windows, clearly the fitting depends greatly on how well the wood is seasoned and how much it shrinks and warps.

The figures do, however, illustrate the point that the amount of heat required to keep a building up to a definite temperature depends very considerably on the *exposure* and the *wind strength*, and on the size and number of windows.

Buildings provided with ventilators and/or fireplaces obviously increase the air interchange even when there is no fire in the grate, because the air has a ready egress up the chimney flue or out of the ventilator. It is, of course, one of the virtues of the open grate that a large and fairly definite quantity of air is discharged from the room every hour, obviously finding its way in from other sources, even though the effect is generally to produce objectionable draughts.

It must be generally admitted that while the heat losses due to radiation can be estimated with considerable accuracy, the losses due to air change, often nearly as important an item, and in some cases even more so, can in most cases only be estimated very roughly. Nevertheless, by using certain constants applied to different types of building built in various ways, figures are obtained which are, at any rate, reasonably consistent among themselves and enable us to say that, as a certain building gave satisfactory results, it may reasonably be assumed that another building calculated on similar lines will also be satisfactory, and this consistency enables us to correlate our experience and make use of it in the future.

Table VII gives, expressed as air changes per hour, a general indication, based on experience, of the minimum infiltration which ought to be allowed in the design of the heating system.

It has occasionally happened that heating contractors, perhaps not those of the highest standing, in designing a heating job in keen competition, have been disposed to minimize the number of air changes and provide a scheme which may be quite inadequate except under very favourable conditions. It is a point of considerable difficulty, and the heating engineer should certainly be entirely free from any financial bias of this kind when estimating what the proper allowance ought to be.

It should also be pointed out that when tests are being made, all doors and windows are to be kept closed. It is, however, obvious that in practice the doors, at any rate, have to be opened from time to time, and in the case of some factories where the products are raw materials which have to be despatched and received, this opening may be almost continuous and cause a very large air change.

It is clearly one of the responsibilities of the heating engineer to esti-

mate the effect of what may be reasonably anticipated, and to make a suitable allowance as a margin over and above what is needed when doors and windows are kept shut.

TABLE VII

NATURAL AIR CHANGES AND ROOM TEMPERATURES

Room or Building	Air changes per hour	Room Temperature deg. F. (based on 30° F. outside)
Assembly Halls	3	60
Banking Halls (Large)	1½	65
Canteens	2	60
Churches and Chapels—Up to 100,000 cu. ft.	1	60
„ „ „ 100,000 to 250,000 cu. ft.	¾	60
„ „ „ 250,000 to 1,000,000 cu. ft.	½	60
„ „ „ Over 1,000,000 cu. ft.	¼	60
Exhibition Halls—Over 14 ft. high	2	60
„ „ Up to 14 ft. high	3	60
Factories—according to type, construction, and occupancy	to 3¾ Sedentary	65
	Light work	60
	Heavy „	55
Flats and Residences—Living Rooms	*1 to 1½	65
„ „ „ Bedrooms	*1½ to 2	55
„ „ „ Entrances, Staircases and Corridors	2	60
Garages (Public)	5	45
Gymnasia	2	55
Hospitals—Wards and Staff Rooms	3	65
„ Dayrooms	3	60
„ Operating Theatres	4	80
„ X-ray Rooms	4	65
Laboratories	4	60
Libraries and Reading Rooms	2	65
Offices	*1½ to 2	65
Storerooms and Warehouses—Storage Spaces	½ to 1	50
„ „ „ Working and Packing Spaces	½ to 1	55
Schools—Classrooms	3	60
„ Assembly Halls	1 to 1½	55
Shops (Small)	3	60
„ (Large)	2	60
Swimming Baths	1½	70
Lobbies, entrance halls generally	4	60
Staircases and Corridors generally	2	60
Cloaks and Lavatories generally	2	55

* The lower air change rate applies where one wall with door or window exposed. The higher rate applies where more than one wall with door or window exposed.

Note: This table does not apply where rooms are mechanically ventilated.

For further information on special types of buildings, see *I.H.V.E. Guide to Current Practice*, from which above data is quoted.

Having established the number of air changes per hour, or the amount of air which will enter the different rooms of a building per hour, the quantity of heat necessary to raise this air to room temperature has to be calculated. This involves no more than multiplying the volume by the specific heat and, of course, by the temperature rise. The specific heat of air may be taken as ·24 B.T.U. per lb., which at 45° F. (an average temperature) is ·019 B.T.U. per cubic foot.†

† Barometric pressure and humidity affect this figure somewhat, but not significantly for ordinary calculations.

Thus, to simplify matters, the cube may be taken and multiplied by the following factors according to the number of air changes selected.

TABLE VIII

Heat required for:		B.T.U.'s per hour per cubic foot per ° F. rise in temp.
½ air change per hour	-	·010
	-	·019
	-	·029
	-	·038
	-	·057
	-	·076

2. TEMPERATURE DIFFERENCE

The sum of the transmission losses and of the infiltration loss—both in B.T.U. per degree difference—requires to be multiplied by the difference between the inside and outside temperature.

The outside temperature is commonly taken at 30° or 32° F. in the south of England, but obviously depends on climate, latitude, etc.* The inside temperature depends on the purpose of the various rooms, and Table VII may be used as a guide to this.

Inner walls or partitions between a warmed and an unwarmed room are taken off separately, and the loss per degree difference multiplied by the appropriate temperature difference. It must be remembered that such walls are generally not subject to wind, so that coefficients somewhat less than the normal values may be allowed.

3. ALLOWANCE FOR HEIGHT AND EXPOSURE

Allowance has to be made for the height of the room, owing to the fact that the temperature which the occupants experience relates to the bottom 6 ft., whereas, owing to the tendency of warm air to rise, the bottom 6 ft. will be at a temperature considerably lower than the average temperature of the room to a degree which increases with its height.

The usual allowance which is made for this factor is given in Table IX.

Fig. 10 gives some temperature measurements taken in a normal storey height, and illustrates the vertical temperature gradients which may be expected with various types of heating, such as radiators under the windows or panel heating in the ceiling. The gradient varies also with the position of the larger cooling surfaces.

Fig. 11 shows this temperature gradient in the case of a north light factory which is heated by low-pressure steam pipes suspended under the roof.

* See also chart of degree-days in Chapter XVI.

It is also necessary to make an allowance for *exposure* in the case of rooms facing north or east, and where wall surfaces are above the general level of surrounding buildings.

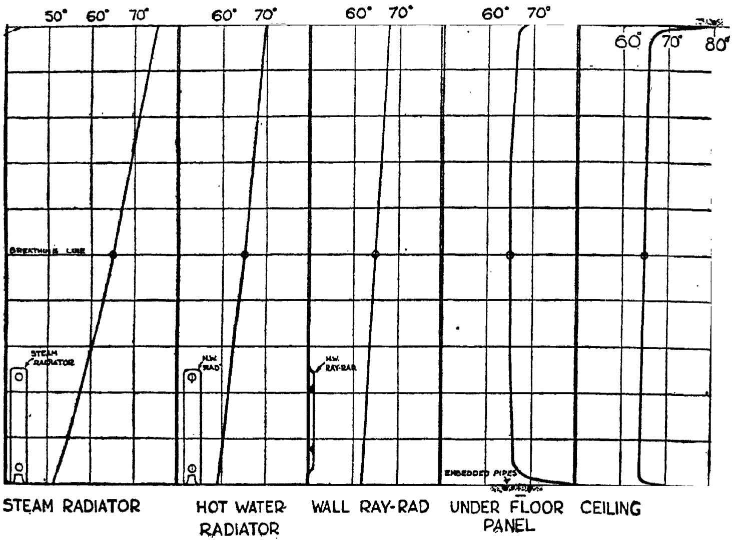


FIG. 10.—Vertical temperature gradient in a room with various types of heating, assuming rooms above and below heated by similar means. See also Fig. 11.

Suitable factors are given in Table IX. They are, of course, purely empirical, and must be applied with discretion to specific cases.

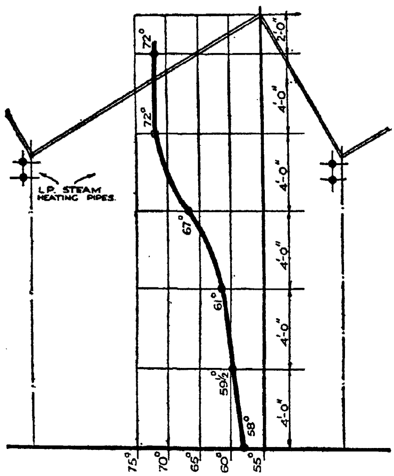


FIG. 11.—Vertical temperature gradient in north light factory.

TABLE IX

HEIGHT AND EXPOSURE FACTORS

HEIGHT

For height of 13 to 17 ft.	add	5%	to total B.T.U. required.
17 ,, 22	,,		
23 ,, 27	,,	15%	
27 ,, 32	,,	20%	
32 ,, 37	,,	25%	
over 37	,,	30%	

EXPOSURE

For N. and N.E. exposure add 10% to wall and window coefficient.

For unusually exposed buildings add 15% to wall and window coefficient.

For 7th floor	and over	add 25%	to total	B.T.U.	required	for each	floor concerned.
,, 6th	,,	,, 20%	,,	,,	,,	,,	
,, 5th	,,	,, 15%	,,	,,	,,	,,	
,, 4th	,,	,, 10%	,,	,,	,,	,,	

4. HEAT LOSSES GENERALLY

Fig. 12 shows a piece of a sheet such as is frequently used for the purposes of making these calculations, filled up for one room. The significance and purpose of the calculation will be appreciated without further explanation.

For the purpose of designing a heating system meticulous accuracy in the calculation of heat losses is, in fact, unattainable as the quality and composition of building materials vary over so wide a range. Further, there is the air change question, which covers so large a proportion of the total heat requirements. It will be noticed, in the example just given, that the heat required to balance the infiltration comes to 820 B.T.U.'s out of a total of 2394 B.T.U.'s per deg. diff.

It must also be remembered that outside temperatures below 30° frequently occur during spells of very cold weather, sometimes for several days and nights on end.

The crucial test of a heating system comes at such periods, and equipment which is well on top of its job and has an adequate margin all round will be found to repay amply the slight extra cost at the start, besides having the advantage of a reduced warming-up time.

Thus, though the heat loss calculations will form the basis of the design of the whole heating system, no matter what form it takes, it is necessary to allow an adequate margin. This particularly applies to an electrical system, where the output is strictly limited by the apparatus installed. In a hot-water system, the temperature in radiators can often be raised above the normal by a little forcing in critical weather.

5. THE VALUE OF BUILDING INSULATION

A glance at the transmission coefficients in Table VI will indicate at once those materials which will give the warmest building, or the ones requiring least heat for the maintenance of the temperature.

*Job No.*__________

Floor.	Room	Air.						Windows and Doors.				Wall.							Roof and Floor.				
		L.	W.	H.	Cube.	Air ch/hr.	Coeff.	L.	H.	Area.	Coeff.	Thickness & Construction	L.	H.	Area.	less w. & d.	Nett Area.	Coeff.	Construction	L.	W.	Area.	Coeff.
																			(SEE DWG.)				
								44	6½	286	1												
								55	2½	138	1												
GD.	WARD	62	20	11	14350	3	·057	7	16½	115	1	11" BR.	129	11	1419	629	790	·27	ROOF	62	20	1305	·32
		13	5					7	6½	45	1	CAVITY								13	5		
										584													
								DOORS 7	6½	45	·75								FLOOR	DO.	DO.	1305	·25
										629													

FIG. 12.—Typical heat loss calculation.

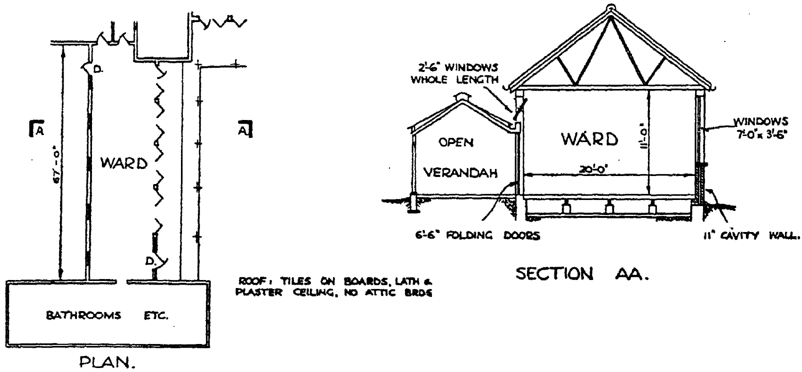
Thus an 11 in. cavity brick wall, plastered, having a coefficient of ·27 will transmit only 50 per cent. of the heat of a 6 in. plastered concrete wall (coefficient ·55). Or a tiled and boarded roof, with plastered ceiling below, having a coefficient of ·32, will transmit only 64 per cent. of 6 in. of concrete plastered (·50).

The old cottage construction of brick and thatch produced very warm buildings. Brick walls two or three feet thick (built when bricks and labour were cheap), plastered both sides and lime washed, and thatch built up in layers, over many generations, made them economical to keep warm in winter and at the same time cool in summer. This type of construction is not practicable for our modern buildings. Thin walls of more conducting materials inevitably tend to depart from all considerations of inherent warmth, except where scientifically corrected.

The typical modern building has a flat roof, and walls of concrete with long horizontal lines of windows. Yet uninsulated concrete and glass are

Sheet No.

B.T.U. for 1° diff.					Temp. Diff.	Height and Exposure Factor	Total B.T.U. per hour	H.S. required @ 160	Pipes				Rads or Panel		Total B.T.U. Provided	Notes
Air	W. & d.	Wall	Roof & Floor	Total					L	Size	Coeff	B.T.U.	Sq. Ft.	B.T.U.		
820	584 33	213	418 326	2394	35	—	84000	525	-	-	-	-	528		84500	



DETAILS OF TYPICAL HOSPITAL WARD.

two of the worst materials that could be chosen from a heat loss point of view, especially as in order to save expense and weight the concrete is generally made as thin as possible.

In the interests of national fuel conservation, the construction of new buildings without insulation should be prohibited.

So far as the glass is concerned there is no means of reducing this loss except by cutting down its area or providing double windows, or a special glass of the 'Thermolux' type, which is, however, completely diffusing. Expense generally rules out the use of double windows, though there is often much to commend them, as the loss is thereby reduced by about half. The walls and roof can, however, be treated with insulating materials such as cork, moler brick, cellular concrete, fibre board, glass silk, aluminium foil, etc.

When considering insulation the inevitable question is—how much will be saved; what shall the material be, how thick, and what will it cost?

TABLE X

COMPARISON OF BUILDING INSULATING MATERIALS

Material	Thick-ness	Cost per sq. ft. fixed (approx.)*	Coefficient of Con-ductivity/sq. ft./deg. diff., for thickness stated/hr.	Relative Insulation Value for given out-lay taking 2 in. cork =100	Position of use
	In.	d.			
Cork	2	7	·15	100	CR, TR, W, FB
"	1	4½	·30	78	
Moler Brick (Solid)	3	6⅔	·20	79	CR
" " "	2	5	·30	70	W
Cellular Concrete (40 lb./cu. ft.)	3	8½	·15	82	CR
" " " "	2	6	·225	78	W
Fibre Board	½	4	·76	35	CR, TR, W, FB
" "	⅝	4	·61	43	
Glass Fibre Quilts	¾	4½	·37	65	TR
" " "	½	3½	·56	54	FB
Aluminium Foil, 'Ardor'	½	6	·21	84	TR, FB, W
Timber, T. & G., Wrot, Deal	1	4½	·87	27	TR, W, FB

Position of Use: CR = On Concete Roof. W = On Wall Surface.
TR = On Ceiling under Tiled Roof. FB = In Timber Frame Building.

* See note on costs in Preface.

Table X has been drawn up to show the relative conductivities of a variety of materials and their costs. For convenience 2 in. thickness of cork, which can be used in practically any position, has been taken as a standard for comparison, though in practice 1 in. cork gives nearly as great a saving with much less expense.

It is therefore proposed to take three typical buildings and see how the heating costs are affected if all walls and roofs are completely treated:

CASE 1 (*a*) *Private House, with* 6 *in. Concrete Walls*

(2 floors of 1200 sq. ft. each at 10 ft. storey height)

Cube	24,000 cu. ft.
Window area	500 sq. ft.
Wall (6 in. concrete plastered) area (net)	2,300 sq. ft.
Roof (4 in. concrete plastered and asphalted) area	1,200 sq. ft.
Floor (board and joist cavity under) area	1,200 sq. ft. (lowest floor only measured).
Air change	1½ changes per hour.
Temperature	65° inside, 30° outside.
Hours of heating per annum	3,000†

Heat Loss per Hour

	Uninsulated. Per deg. diff.		Insulated Walls and Roof, 1 in. Cork. Per deg. diff.
Air change, 24,000 × ·029	680		680
Windows, 500 × 1·0	500		500
Wall, 2,300 × ·55	1,265	× ·20	460
Roof, 1,200 × ·58	695	× ·21	252
Floor, 1,200 × ·30			360
	3,500 B.T.U.'s per hour		2,252 B.T.U.'s per hour

† From Table LXIX, p. 398.

For 35° rise	3,500 × 35° = 122,000 B.T.U.'s per hour		2,252 × 35° = 79,000 B.T.U.'s per hour
Per annum	122,000 × 3,000 hours* = 366,000,000 B.T.U.'s = 3,660 therms per annum		79,000 × 3,000 hours = 237,000,000 B.T.U.'s = 2,370 therms per annum
Coal or coke at 40s. per ton = 3·3d. per therm* (100,000 B.T.U.'s)	£50 per annum	Saving £17	£33 per annum
Electricity at ½d. per unit = 14·6d. per therm; allowing saving due to thermostatic control 25 per cent.	£167		

The cost of 1 in. thickness of cork insulation in this case will be approximately:

3,500 sq. ft. at 4½d.	£66
The saving in cost of the heating system, whether coal or electric fuel, might be	£50
Net extra cost of insulation	£16

It will be seen that with coal or coke fuel the extra cost of insulation is paid for out of savings in 1 year; this is surely adequate justification for its adoption. With electricity the savings are even more marked.

CASE 1 (*b*)

If the walls were of brick and the roof of tiles and boarding with plaster ceiling, a similar calculation shows that with the same building and cost of fuel the savings will be:

	Per annum.
With coal or coke	£11
With electricity	£35

The net cost of insulation would be about £50, so that with cheap fuel it would be repaid in 4½ years and with electricity in just over 1 year.

CASE 2 (*a*) *Office Building*

(5 floors at 10,000 sq. ft. each × 10 ft. storey height)

Cube	500,000 cu. ft.
Window area	7,000 sq. ft.
Wall (6 in. concrete plastered) net area	18,000 sq. ft.
Roof (6 in. concrete plastered and asphalted)	10,000 sq. ft.
Ground floor (wood block on concrete)	10,000 sq. ft.
Air change	2 changes per hour.
Temperature	65° inside, 30° outside.
Hours of heating per annum	1,600*

Heat Loss per Hour

	Uninsulated. Per deg. diff.		Insulated Walls and Roof, 1 in. Cork. Per deg. diff.
Air change, 500,000 × ·038	19,000		19,000
Windows, 7,000 × 1·0	7,000		7,000
Wall, 18,000 × ·55	9,900	× ·2	3,600
Roof, 10,000 × ·5	5,000	× ·2	2,000
Floor, 10,000 × ·15	1,500		1,500
	42,400		33,100
For 35° rise	35		35
	1,480,000 B.T.U.'s per hour		1,160,000 B.T.U.'s per hour
	1·48 million × 1,600 = 23,700 therms		1·16 million × 1,600 = 18,600 therms

* From Tables LXVII and LXIX, pp. 393–8.

Coal or coke at 40s. per ton = 3·3 per therm*	£326	£25 Saving £70 per annum.
Electricity at ½d. per unit = 14·6d. per therm; allowing saving due to thermostatic control 25 per cent.	£1,080	Saving £240.

The cost of 1 in. thickness of cork insulation in this case will be approximately:

28,000 sq. ft. at 4½d.	£525
The saving in cost of the heating system, whether coal or electric fuel, would be about	£450
Net extra cost of insulation	£75

Thus the initial cost of insulation is saved in a year or less in both cases.

CASE 2 (*b*).

If the same building has 13½ in. brick walls and concrete roof, the savings are £50 per annum with coal, and £175 with electricity on a net extra outlay of £300 for insulation.

CASE 3 *Factory Building, Framed Construction*

(300 ft. × 100 ft. × 20 ft. to eaves, 25 ft. mean)

Cube	750,000 cu. ft.
Windows	2,000 sq. ft.
Roof glass	6,000 sq. ft.
Corrugated asbestos sheeting walls and pitched roof, less glass area	45,000 sq. ft.
Floor, concrete on earth	30,000 sq. ft.
Air change	1 per hour
Temperature	60° inside, 30° outside
Hours of heating per annum	1,600*

Heat Loss per Hour

	Uninsulated. Per deg. diff.		Insulated Walls and Roof, ⅝ in. Fibre Board or equivalent
Air change, 750,000 × ·019	14,200		14,200
Windows 2,000 × 1·0	2,000		2,000
Roof glass 6,000 × 1·1	6,600		6,600
Sheeting 45,000 × 1·4	63,000	× ·31	14,000
Floor 30,000 × ·13	3,900		3,900
	89,700		40,700
Height factor 10%	9,000		4,000
	98,700		44,700
For 30° rise	2,961,000 B.T.U./hr.		1,341,000 B.T.U./hr.
Per annum 1600 hrs.*	47,400 therms		21,400 therms
Coal or coke at 40/- a ton (3·3d./therm)*	£652 per annum		£296 per annum
		Saving £356 per annum	

Cost of Insulation, 45,000 sq. ft. at 4d.	£750
Saving in cost of heating system approximately	£400
	£350

In this case the saving covers the extra initial cost in approximately one year.

* Taken from Tables LXVII and LXIX, pp. 393–8.

TABLE XI

SAVINGS DUE TO BUILDING INSULATION

Type of Construction and Insulation.	Cost of Insulation after deducting saving on heating system.	Saving of Fuel per annum.			
		With coal or coke at 40/- ton.	Years for repayment.	With electricity at ½d. per unit.	Years for repayment.
Case 1. Private House, 24,000 cu. ft.					
Walls, 6 in. Concrete; Roof, 4 in. Concrete; Both with 1 in. Cork	£16	£17	1	£59	·27
Walls, 9 in. Brick; Roof, Tiles and Boards; Both with 1 in. Cork	£50	£11	4·5	£35	1·4
Case 2. Office Building, 500,000 cu. ft.					
Walls, 6 in. Concrete; Roof, 6 in. Concrete; Both with 1 in. Cork	£75	£70	1	£240	·3
Walls, 13½ in. Brick; Roof, 6 in. Concrete; Both with 1 in. Cork	£300	£50	6	£175	1·7
Case 3. Factory, 750,000 cu. ft.					
Sheeted Walls and Roof lined Fibre Board	£350	£356	1	—	—

Table XI summarizes the above conclusions. The effect on savings and first cost of substituting other materials for cork can easily be estimated by using the appropriate factors for the insulation used.

Various methods of using these materials are illustrated in Figs. 13 and 14, pp. 36 and 37.

Other Considerations in Insulation of Buildings—Another important effect of insulation on the inner surface of a wall or ceiling is the reduction which it effects in the time necessary for heating up. It has been shown by test* that a room lined with timber panelling and heated by a gas fire requires one hour for a temperature rise from 47·5° to 62°, as compared with over two hours untreated. A similar effect is brought about by any insulator used as a lining.

A further advantage of insulation is in the summer. A 6 in. uninsulated concrete roof, subject to the sun's radiation, will, after a few hours, arrive at a temperature of about 80° to 85° on the underside. If 1 in. cork or 2 in. moler brick is introduced on top of the roof under the asphalt it can be shown that this temperature is lowered by about 10°, or very little above the inside air temperature. This means that insulation of the ceiling will prevent its acting as an effective panel heating system, which is what it is really doing, and which, to say the least, is not generally welcomed in summer.

*Dufton, 'Tests carried out at B.R.S., Watford'. See *Phil. Mag.*, 1931, p. 1233.

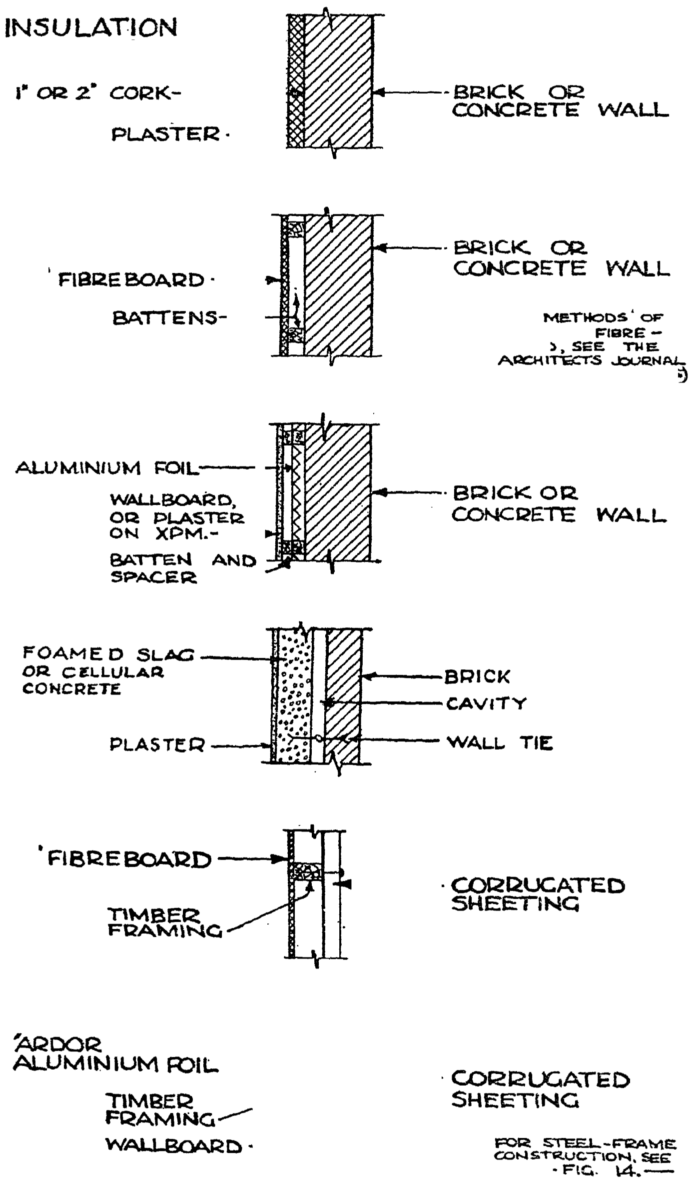


FIG. 13.—Methods of applying insulation to walls.
(For transmission coefficients see Table VI.)

A simple, cheap and effective roof insulator is washed shingle three or four inches deep, placed loose on the top surface. Experiments made by the Building Research Station show that a 2 in. layer of gravel on a 7 in.

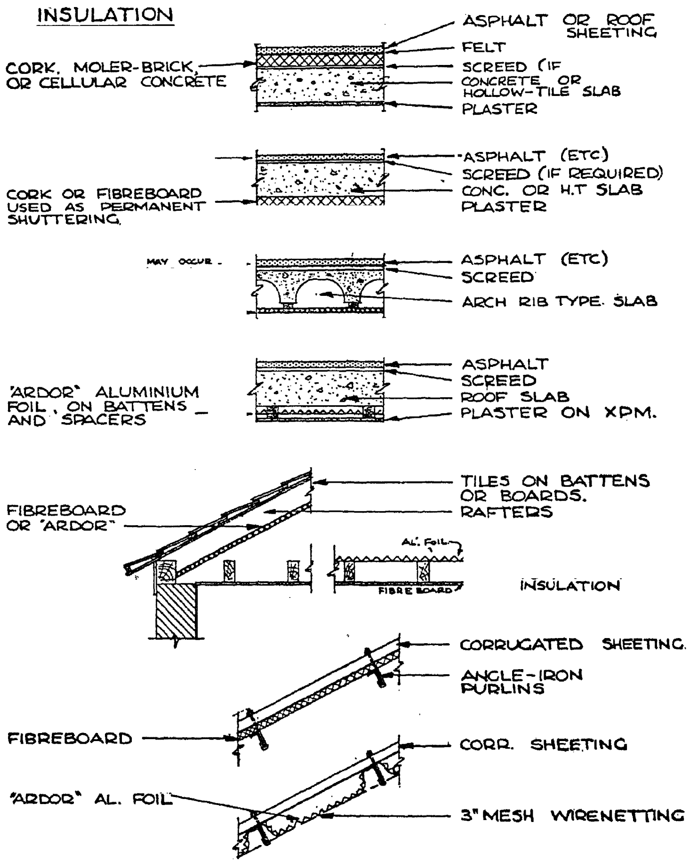


FIG. 14.—Methods of applying insulation to roofs.
(For transmission coefficients see Table VI.)

concrete roof-slab may reduce the top surface temperature by as much as 27° in the sun during hot summer weather.

Reference has been made (p. 13) to the advantage of a white surface for reducing the sun heat gain when applied to a roof. Various proprietary methods of providing such a surface are available, mostly in tile form, and in some cases hollow so as to introduce an air gap.

Where the ground floor of a building heated by ceiling panels has a timber floor with ventilated cavity under, the radiation impinging on the floor is largely lost to the cold air below. For this reason it is advisable in such cases to insulate the underside of the floor boards.

It is not suggested that the methods of insulation described above cover the whole range of available materials, though the principal ones have been referred to. Certain materials, such as compressed peat, sawdust, straw, grain husks, though effective insulators, tend to encourage vermin and are inflammable; slag wool suffers from the disadvantage of being an unpleasant material to handle, though it is now available in improved forms.

6. CONDITIONS OF COMFORT

The question arises, what is the heating system for: is it to provide a specified temperature in a building or is it to furnish conditions of comfort? Many would give the answer, 'The one involves the other'; but there is more in it than this.

The calculations so far described are based on the former method, that is, it is assumed at the outset that a uniform temperature, as measured by an ordinary thermometer, of 60° or 65° is necessary. Yet it has already been pointed out that the mercury thermometer does not properly take into account the whole question of comfort.

The subject may now be approached from another standpoint. The human body is one which emits heat at a given rate by radiation, convection and evaporation. If placed in cold surroundings, that rate will increase, though this is to some extent corrected physiologically by the closing of the pores of the skin and by the removal of the blood from the outer tissues. Nevertheless there is an increased heat output and a feeling of chilliness.

In such cold surroundings the body may, however, be brought back to a state of comfort by allowing heat radiation to fall upon it. Thus, by turning on an electric bowl fire in a cold room an instantaneous feeling of comfort may be brought about by standing in the beam. As the air in the room gets warmer, so a lesser intensity of radiant heat is required for comfort. The converse effect also holds good, that in a room with cold walls, etc. (i.e. cold radiation), a state of comfort can be achieved by warming the air to a sufficiently high temperature. A state of equilibrium is established whereby the heat lost from the body by radiation to surrounding objects and by convection to the air is equalized by the absorption of heat from an outside source.

Thus for every air temperature there is a corresponding rate of radiation from other objects or surfaces, either warm or cold, which will produce a state of comfort.

To give a basis for estimating comfort conditions there has been devised a conception, known as *Equivalent Temperature*. This attempts to combine the effect of air temperature, radiation and air movement (but not humidity).

One method of measuring the Equivalent Temperature is by the

*Eupatheoscope.** We are indebted to Mr. A. F. Dufton, M.A., D.I.C., for the following explanation of this instrument:

'The temperature of a room with air and walls at different degrees is not easily specified. From the point of view of comfort it is the rate at which heat is lost from the body which seems to be important, and the *Equivalent Temperature* of an environment is defined as that temperature of a uniform enclosure in which, in still air, a sizeable black body would lose heat at the same rate as in the environment, the surface of the body being one-third of the way between the temperature of the enclosure and 100° F.

'The scale of equivalent temperature is essentially a refinement of the sensation scale. In terms of our sensations we may say, "It is colder now that the sun has gone in", or "Come round the corner, it will be warmer out of the wind". But when we attempt to express "how much colder" or "how

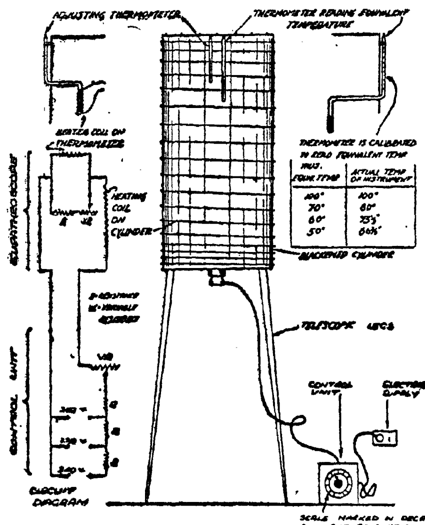


FIG. 15 (*a*).—The Eupatheoscope.

much warmer", we need to have a scale of temperature. Human sensations are affected by diurnal and seasonal acclimatization and are notoriously variable and individual. For accurate work, therefore, physiological sensations must be interpreted in terms of physical quantities capable of exact measurement.

'Figure 15 (*a*) shows a *Eupatheoscope*, an instrument for measuring equivalent temperature, which was designed primarily as an instrument for research. When the instrument is placed in a room with air and walls at, say, 62° F., and the air is still, it indicates 62° F. That is the equivalent temperature. If there is a draught of some 20 ft. per minute in the room,

* The description applies to the new version. The previous instrument measured heat input for constant surface temperature.

the instrument feels colder and indicates 56° F. Although the room is uniformly at 62° F., it is effectively 6° colder because of the draught.'

Other methods of combining these factors in a single instrument or scale are:

(*a*) Dr. Vernon's *globe thermometer** has already been mentioned as giving a useful guide to the effect of radiation over ordinary air temperatures. It is much more convenient than the eupatheoscope, but its value with air movements is doubtful, and it does not take humidity into account.

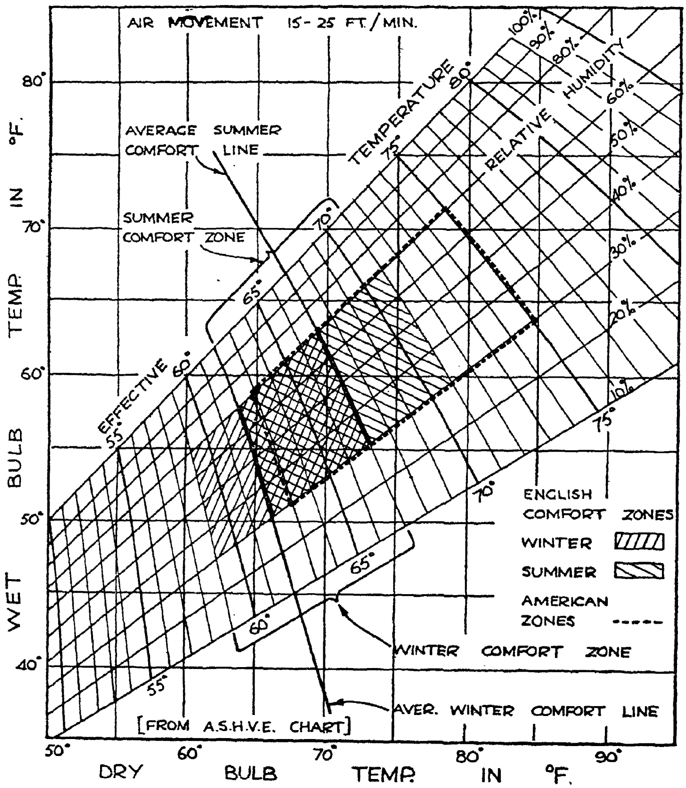


FIG. 15.—'Effective temperature' chart, showing winter and summer comfort-zones applicable to Great Britain only. The chart takes no account of radiation, and its use is limited to rooms, offices, homes and the like, where the occupants remain long enough to become accustomed to the conditions—say three hours' occupancy. The boundary of the corresponding zones recommended for use in the United States is shown by a dotted line.

(*b*) *The Kata thermometer* is an instrument consisting of two thermometers, having wet and dry bulbs of fair size filled with spirit. The stems are graduated only between 95° F. and 100° F., i.e. over the range approxi-

* Dr. T. Bedford, *Journal I.H.V.E.*, 1934-5, p. 544.

mating to body heat. The bulbs are plunged in hot water so as to raise them above 100° F., and the time taken for each to cool between the two temperatures named is noted with a stop-watch. The dry bulb is, of course, dried after removal from the hot water. The wet bulb, being covered with a muslin sheath, remains wet.

The dry bulb cooling time is dependent on heat loss by radiation and convection, and may be used with a calibration chart to give air velocities, particularly at low speeds. From this may be deduced the freshness or stagnancy of the atmosphere of a room. The wet bulb is not in a condition representative of the human body, and is therefore of little help in comfort problems.

(*c*) In the United States an *Effective Temperature* scale has been devised combining air movement, air temperature and humidity. This scale is the outcome of a long series of tests carried out on a number of people in a great variety of conditions. They were required to register their feelings, and from the results a series of curves, known as effective temperature lines, were drawn, from which a zone of comfort could be established. This scale (see Fig. 15) is very thorough and complete, but does not include the effect of radiation. The shaded zones in the figure are for this country and are somewhat cooler than the corresponding American ones, since the English idea of comfort is rather different from the American.

Examination of Fig. 15 shows that an effective temperature of 60° F. may be satisfied by either of the following conditions:

60° F. *Effective Temperature*

Dry bulb temperature -	- 60° F.	Dry bulb temperature -	
Wet " " -	- 60° F.	Wet " " -	
Relative humidity - -	- 100%	Relative humidity - -	

Similarly an effective temperature of 75° F. may be satisfied by either of the following:

75° F. *Effective Temperature*

Dry bulb temperature - -	- 75° F.	Dry bulb temperature -	F.
Wet " " - -	- 75° F.	Wet " " -	F.
Relative humidity - - -	- 100%	Relative humidity -	10%

It will be seen that the effective temperature lies between the wet and dry bulb temperatures.

How then is the criterion of comfort to be established? The various ingenious methods described above are not, as will be appreciated, a complete solution, nor has there yet been devised an instrument sufficiently simple for practical and general use, yet accurate in its relationship to the standard of human comfort.

We find, therefore, that the ordinary thermometer still remains as the practical standard of determining whether a building is properly heated or not, the actual temperature required being somewhat affected by humidity, as in the Zone of Comfort Chart. And, after all, the thermometer has

the great merit of simplicity, since it is difficult to persuade the average Englishman that his office is comfortable when his thermometer stands at 55°, however much one may argue that this deficiency in air temperature is made up by radiant heat.

Radiant heat in a closed cold room sooner or later raises the air temperature by warming the objects on which it falls, and they in turn warm the air by convection. The air temperature and radiant temperature will thus tend to approach one another, and probably the most comfortable condition exists when they are nearly equal, with the air slightly cooler than the mean radiant temperature. This condition is one in which the effect of radiation plays so little part that the readings of an ordinary thermometer is probably a safe standard of comparison, but it does not hold good if the radiation is from a high temperature source.

7. INCIDENTAL SOURCES OF HEAT

The purpose of heat loss calculations is, therefore, clearly to form a basis for determining the amount of heat to be liberated in a room so as to maintain a given temperature as measured by an ordinary thermometer.

This being the case, what is the effect of occupants or other heat-producing agents being placed in the room?

Allowance for Occupants—Heat is emitted from the human body, partly as sensible heat and partly as latent heat in the sweat and vapour in the breath. The sensible heat alone affects the temperature of the room.

Table XII gives average values at different states of activity in an atmosphere at 65° F. and 60 per cent. relative humidity.

TABLE XII

HEAT FROM OCCUPANTS

	Heat dissipated B.T.U.'s/hr. per Occupant		
	Total	Latent	Sensible
At rest	400	100	300
Sedentary worker	650	250	400
Walking	900	400	500
Light manual work	1,200	600	600
Heavy " "	1,600	800	800

The sensible heat dissipated is reduced at higher temperatures, becoming zero at 98° (blood temperature), and increases at low temperatures (e.g. at 32° about 70 per cent. greater than at 65°). The latent heat follows the opposite course, decreasing at lower temperatures and increasing at higher.

The office building containing 50,000 sq. ft. of floor previously considered (p. 33) might, if closely packed, accommodate 2500 people (at 20 sq. ft. each). These could produce $2500 \times 300 = 750{,}000$ B.T.U.'s per hour. This represents three-quarters of the total heat requirements for the insulated building (see previous calculations) with 65° F. inside and 30° F.

outside. In warmer weather the 2500 occupants will raise the temperature above 65°, with no other heating at all.

Allowance for Machines—Energy expended by machines within a building results for the most part in the production of heat. In a lathe, for instance, in which metal is turned, the heat liberated is divided between the tools, the metal and the lubricant, which subsequently dissipate their heat in the room. The friction losses of shafting, belt drives, etc., are converted into heat. An electric motor driving machines transmits dynamic energy perhaps to a number of points within a building, where each machine in turn converts it into heat. The efficiency losses within the motor itself similarly come out in the form of heat. The exception is where work is done to produce potential energy, as in the pumping of water to a height, or in the lifting of goods in a hoist; in such cases similarly the losses are converted into heat.

The heating effect of work done inside an enclosure is the product of the power consumed and the mechanical equivalent of heat. Thus, if 50 H.P. is expended the heat released will be $50 \times 2544 = 127200$ B.T.U.'s/hr. If the motor is in the same room as the machine and consumes 45 K.W., 8 K.W. may be wasted in inefficiency and 37 K.W. converted into useful energy; the heat liberated in the room will be the total

$$45 \text{ K.W.} \times 3415 = 153{,}600 \text{ B.T.U.'s/hr.}$$

Allowance for Light—Electric light may be treated as machinery. Thus, in the office buildings previously referred to, 50,000 sq. ft. of floor may require 50,000 watts of lighting. This produces $50{,}000 \times 3{\cdot}415 = 170{,}750$ B.T.U.'s per hour, i.e. about one-sixth of the total winter heat loss or enough to raise the temperature about 6°. With gas lighting about seven times as much heat is liberated to produce a given amount of light as compared with electricity.

Allowance for Process Equipment—The liberation of heat within the enclosure from equipment such as steam presses, hot plates, drying ovens, gas or electric furnaces, etc., may be determined by multiplying the hourly consumption with the heat value of the unit in question: e.g. electrical unit, 3415 B.T.U.'s/unit; gas, 100,000 B.T.U. therm; steam, 1000 B.T.U.'s/lb.

Use of Allowances—Heat gains from any of the above sources, occupants, machines, lighting, and other equipment, cannot usually be relied upon to relieve the heating system of any part of its work as they are often intermittent and variable. Thus, it is usual to ignore them in determining the heat input necessary to maintain the required internal temperature; in practice they reduce fuel consumption. In the design of ventilation or cooling plants these heat gains are of the highest importance and are dealt with in a later section.

Allowance for Sun—While the heat from the sun absorbed by a building is most important when considering cooling plants under summer conditions (and is treated later in Chapter XVIII), it is so unreliable in winter

that no allowance can be made for it when designing a heating plant, as in the coldest weather it may be entirely absent.

Nevertheless, there are other days, even in winter, when its effect on the southerly aspect may be quite important, while rooms facing north receive no such benefit. Hence, in important installations, there is much to be said for having rooms with south aspect on a separate set of mains, so that they can have heat reduced without affecting that in rooms facing north.

8. TEMPERATURE CONTROL

From what has already been said, it is clear that the ideal system is one in which each room has separate automatic control. It is then supplied with a heating element sufficient to warm it through the full range of temperature without allowing for:

(*a*) heat from occupants,	(*c*) heat from light,
(*b*) ,, ,, machines,	(*d*) ,, ,, sun,

and the heat is automatically reduced as and when these factors operate, which may be at quite different times and in different degrees in all the rooms of a building.

Thus, a room facing south may be sufficiently warmed by the sum of (*a*), (*b*) and (*d*), while a neighbouring room facing south might be empty of occupants and machines and receiving only heat from the sun, while a third room facing north might receive no heat at all from any of these sources. Under these circumstances, the first room might need its heat cut off completely, the second one one-half, and the third one left full on.

A system which automatically allows for all such combinations is the ideal, other things being equal. Any departure from this ideal involves loss of comfort and waste of heat through the opening of windows in rooms which are otherwise too hot. In practice, a compromise is generally to be effected between the ideal, which may be expensive in first cost, and the primitive, which may be expensive in fuel. The more costly the fuel, the more important this item becomes.

Systems of direct electric radiation are perhaps those where the fuel cost is the highest, and these can be so controlled as to take the fullest advantage from complete automatic local regulation. In this way as much as 50 per cent. of the heat may be saved, though the capacity of the installation cannot be reduced. It is, however, only fair to note that even with 50 per cent. saving, the electric running cost will generally be higher than the running cost of a hot-water system on solid or liquid fuel. Indeed, there is no reason why the latter should not also have complete automatic local control, as a thermostat in every room can be supplied under either system. Thermostatic control to this extent is, however, expensive. The interest and depreciation on complete individual control of rooms will often be found to exceed any possible saving in fuel, and its value has then to be considered on the grounds of comfort and convenience.

CHAPTER III

Considerations affecting the Choice of Heating System

The useful heat required for each room or portion of the building having been arrived at as described in Chapter II, it is now necessary to consider how this heat shall be supplied. It should be noted that whereas the useful heat required depends entirely on the characteristics of the building, i.e. on its size, construction, exposure, air change, etc., the total heat to be supplied must also include allowance for all the losses which may occur between the generation of the heat and its dissipation in the room to be warmed. This in turn depends on the system employed, the best choice for any given building being that in which the greatest proportion of heat generated is usefully employed.

All heating systems may be divided into two groups:

(*a*) 'Direct' or 'Local' systems, such as gas or electric radiators, in which the fuel or energy purchased is consumed in the room to be heated.

(*b*) 'Indirect' systems, usually using hot water or steam, in which the energy is consumed at some more or less centralized point outside the room to be heated.

This classification is discussed at greater length in Chapter IV, but it is convenient here to consider briefly some other points bearing on the choice.

1. LOSSES IN MAINS

It is clear that with an indirect system the hot water, steam or other heating medium must be transmitted through pipes to the room to be heated, and that some heat will be lost from these pipes, depending in amount on their size, length and insulation.

In a compact system the mains may be short, so that, when well insulated, the loss may amount to as little as 10 per cent. Indeed, in some cases, the greater part of the mains may be exposed in rooms to be warmed, where their radiation and surface is useful, and does not constitute a loss.

In straggling jobs, on the other hand, the mains may consist of large pipes running hundreds of yards in underground corridors or tunnels where the mains loss may frequently be 30 per cent., even when insulated. Here, of course, the insulation efficiency becomes very important, both for first cost and fuel cost. In the extreme cases, it becomes an important factor in deciding between a *central system* and a *decentralized* one.

By a *central system* is meant one with a single boiler house or its equiva-

lent, and by a *decentralized* system, one with more than one boiler house or its equivalent. This is not to be confused with a *local* system, which has a heat-generating unit (such as a gas fire) in each individual room.

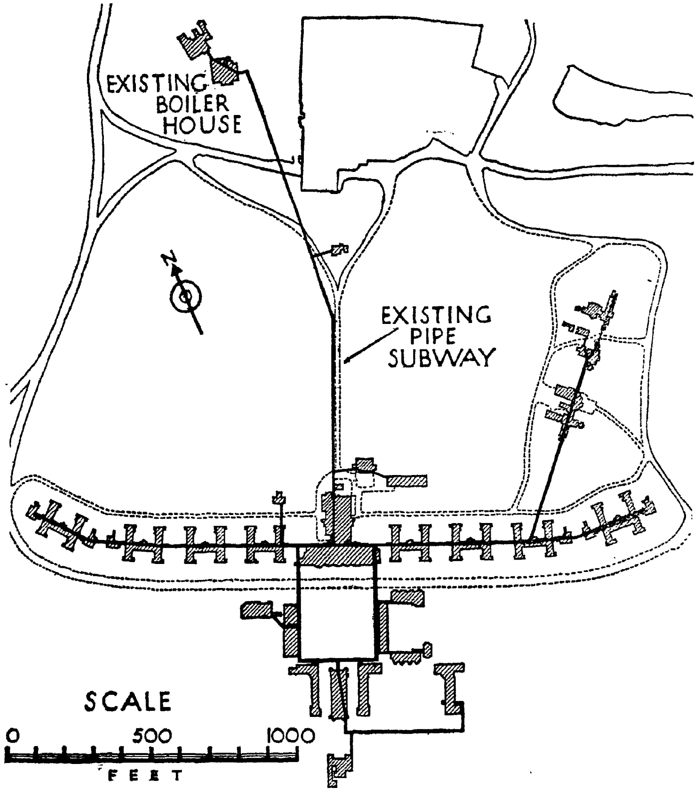


FIG. 16.—Block plan showing mains at Christ's Hospital, Horsham.

As against the mains losses, the advantage of the central system may be summarized as follows:

(1) Less labour and supervision.
(2) Gives a better diversity factor.
(3) Can burn cheaper fuel and take advantage of mechanical stokers.
(4) Requires only a single flue, boiler house, pump, etc.
(5) Less loss on boiler radiation.
(6) May enable an economic combination with electric generating plant to be arranged.

In large institutions a completely centralized system is generally to be advocated. It may be of interest, by way of example, to mention that at

Christ's Hospital (see Fig. 16,), the system was reorganized a few years ago from a partly decentralized to a completely centralized one.

Under the old conditions steam was supplied to calorifiers in a number of the buildings from the main boilers, but in addition there were upwards of a dozen separate boilers for heating and hot-water supply in the isolated blocks. Altogether there were about 50 separate heating units of various types, each requiring attention.

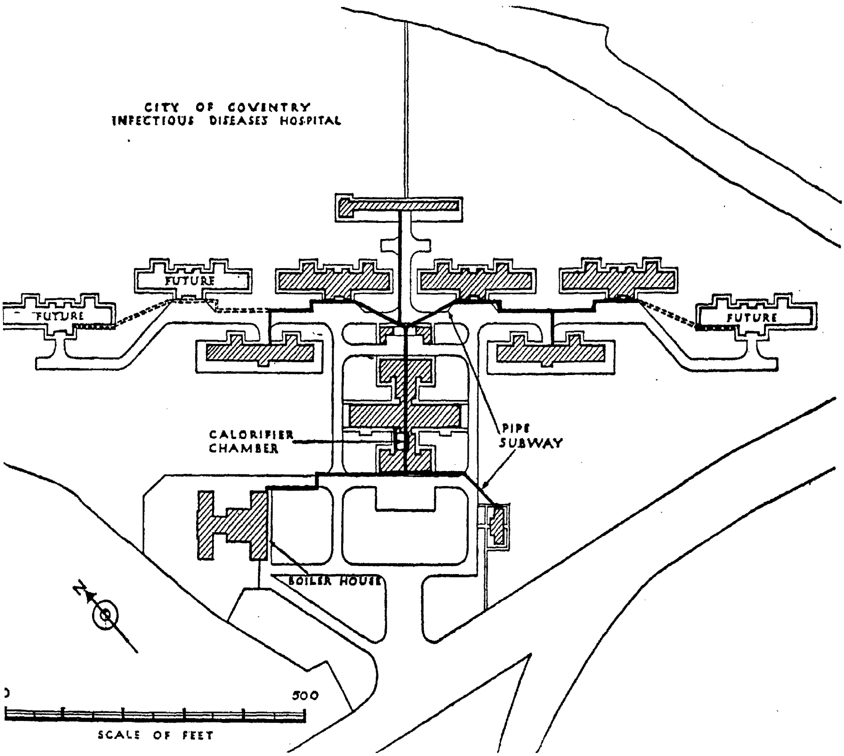


FIG. 17.—Block plan showing mains at Coventry Infectious Diseases Hospital.

By centralization of the heating and hot-water supply systems, all these were replaced by four large calorifiers in the main boiler house, under the care of one man. Coal at 23s. a ton sufficed for the main plant, whereas for the smaller units a fuel costing 40s. a ton had to be used.

The labour of one man was saved, and more heat and hot water were supplied than formerly, giving much more beneficial results.

A further important advantage was that the exhaust steam from the electrical generating engines was utilized in the centralized calorifiers for heating the hot water, whereas previously it had been largely wasted. Altogether a fuel saving of about £1000 per annum was made with the

reorganization, in spite of the mains losses, which amounted to some 25 per cent. of the total heat supplied.

Plate II (facing page) illustrates the mains in one of the subways.

Another example of a centralized system is the Coventry Isolation Hospital shown in Fig. 17. Here also it was clearly proved that a completely centralized scheme possessed the balance of advantages.

There are many examples of extensive systems covering sometimes as much as 2000 acres, where mains generally operating with steam serve factory estates, camp sites, hostels, and so on.

In America and on the Continent, development in the direction of centralization has gone so far that in New York alone there are about seventy miles of heating mains in the streets feeding adjacent buildings, with a heat capacity of 6000 million B.T.U.'s per hour. Such systems are known as *District Heating*; but this is a separate subject. It is unnecessary to go into this in further detail here, as the point has perhaps been made that centralization pays, even when applied to quite extensive and distant groups of buildings.

Among the advantages of the centralized system was mentioned 'good diversity factor'. This applies more to hot-water supply than to heating. It means, in effect, that the coincident peak load for a number of buildings will be less than for one block by itself. Thus a centralized system can often have a smaller plant capacity than the total necessary for each building independently and yet give a greater reserve at each point.

The advisability of using separate sub-circulating systems from separate calorifiers in each building, fed by steam from a central boiler house, depends on the relative distance between the blocks, the size of the blocks, cost of calorifier chambers and whether private generation of electricity forms part of the scheme. In general it may be said that the separate calorifier system with heat distribution by steam or high pressure hot water is preferable on more extensive sites, and the circulation of hot water direct from a central source on the less extensive.

2. LOSSES IN BOILER OR OTHER HEATING UNIT

In addition to the losses from mains, there will be losses in the boiler house itself, so that all the energy in the fuel is not passed into the system. These losses include:

(i) Radiation from Boiler Surface—In large well-insulated installations, this loss may amount to as little as 1 per cent. of the heat received by the boiler. In small badly-insulated installations the figure may rise to 20 per cent.

Actual figures taken from a few examples are as given in Table XIII.

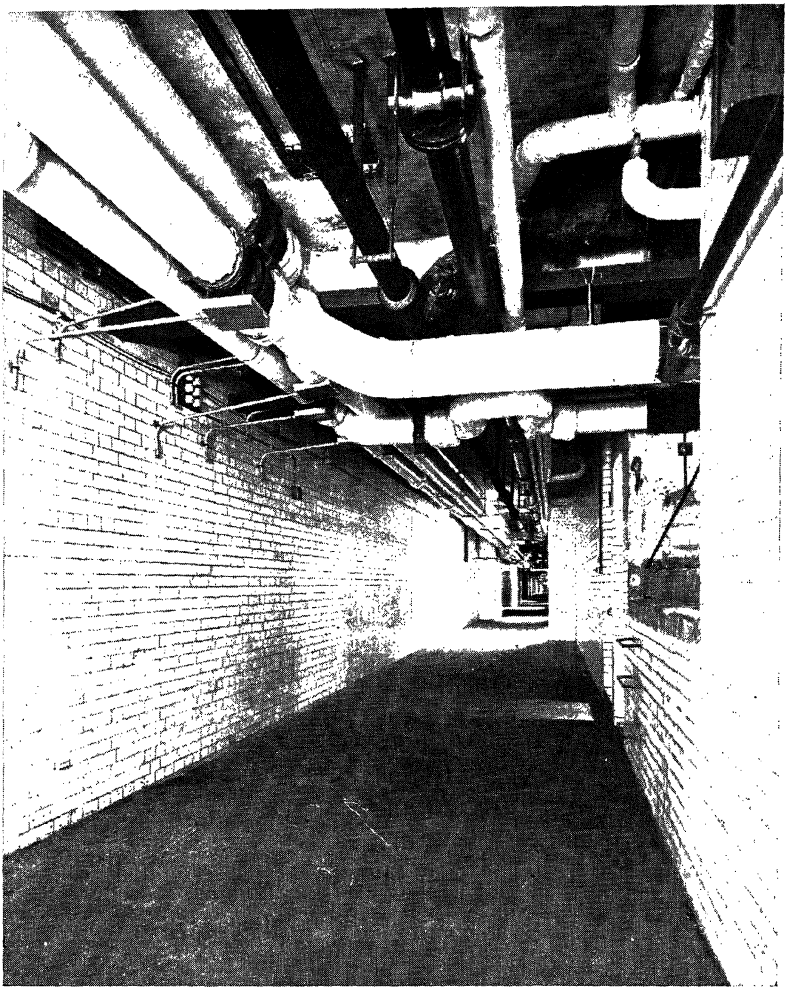


Plate II. Typical example of heating, hot water supply and steam mains from a centralized plant. The main subway at Christ's Hospital (see p. 48)

Plate III. Example of Panel Heating, Bank of England. The pipes are laid on the shuttering before the concrete is placed. Note the ribbed rubber sheeting to give a key for the plaster, and also the fixing of reinforcement over the pipes, commenced in extreme foreground. The pipes in this instance are copper (*see p. 184*)

TABLE XIII

BOILER RADIATION LOSSES

Size and Type of Boiler	Rating	Steam or Hot Water	How Insulated	Boiler Radiation Loss
Cast Iron, Sectional	1,100,000 B.T.U./hour	H.W. 150° F.	1 in. Magnesia, ½ in. Hard-Setting	4%
Welded Steel (Metropolitan), 8 ft. × 7 ft. × 8 ft.	2,500,000 B.T.U./hour	H.W. 180° F.	Aluminium Foil and Steel Jacket	
Electrode Boiler and Cylinders	3,000,000 B.T.U./hour	H.W. 250° F.	3 in. Cork	
Double Return Tube Economic, 8 ft. × 6 in. dia., × 12 ft. 0 in. long	9,000,000 B.T.U./hour	H.W. 150° F.	1½ in. Magnesia, ½ in. Hard-Setting, Polished Aluminium Casing	0·7%
Lancashire, 9 ft. 0 in. dia. × 30 ft. 0 in. long with Economiser	12,000 lbs./hour	Steam 150 lbs./sq. in.	2½ in. Magnesia, ½ in. Hard-Setting, Brick Set	4·7%
Water-Tube, Tri-drum	50,000 lbs./hour	Steam 120 lbs./sq. in.	Brick-Set. Drums Insulated. 2½ in. Magnesia. ½ in. Hard-Setting	1·8%

(ii) Losses in Combustion and Flue Gases—These losses depend on the following factors:

(*a*) Size and type of boiler.
(*b*) Output and degree of forcing.
(*c*) Control of air supply.
(*d*) Method of firing.
(*e*) Fuel.

It may be said here that under the best conditions a boiler rarely has a loss of less than 10 per cent. with gas, and 15 per cent. with coal, coke or oil, and these figures (90 per cent. and 85 per cent. efficiency respectively) are only attained under best combinations of conditions.

With hand stoking, these conditions are difficult to maintain except for short periods, and probably 25 per cent. to 30 or even 40 per cent. losses are more usual in average cases on continuous working. In small domestic boilers, forced, where the flames go straight up the chimney, even greater losses result.

Typical test efficiencies in a few good examples are given in Table XIV.

TABLE XIV

EFFICIENCIES OF SOME TYPICAL BOILER INSTALLATIONS UNDER TEST CONDITIONS

Size and Type of Boiler	Steam or Hot Water	Nature of Fuel	Flue-Gas Temperature deg. F.	CO_2 per Cent.	Boiler Efficiency per Cent.
Cast Iron Sectional	H.W.	Coke	500°	7-11·5	71
Cast Iron Sectional with Automatic Stoker	H.W.	Coal	500°	11-14	75
Welded Steel Gravity Feed	H.W.	Coke	500°	14-15·8	77
Gas Boiler	H.W.	Gas	250°	15	90
Double-Return Tube Economic 8 ft. 6 in. dia. × 12 ft. 0 in. long	H.W.	Oil	250°	7-10	89
Brick-Set Economic, 8 ft. 0 in. dia. × 14 ft. 0 in. long	Steam 125 lbs./sq. in.	Oil	400°	8-11·5	76
Lancashire, 9 ft. 0 in. dia. × 30 ft. 0 in. long with Economiser	Steam 150 lbs./sq. in.	Coal	460°	11·5	79
Water-Tube, 50,000 lbs./hour	Steam 120 lbs./sq. in.	Coal	390°	9	83
'La Mont' Water-Tube, 12,000,000 B.T.U./hour	High Pressure, H.W., 80 lbs./sq. in.	Coal	520°	11·5	80

It will be appreciated that an electric thermal storage system has radiation and mains losses in common with any hot-water circulating system, of which it is only a special type, but has no loss from combustion and flue gases. This loss, indeed, was already incurred when the fuel was burnt at the generating station, and has already been paid for in the price of electricity.

3. SUMMARY OF LOSSES AND ESTIMATION OF BOILER CAPACITY

The heat-producing apparatus whether boiler, calorifier or electric heater, must be capable of supplying the total heat requirements of the system as determined from the heat losses, plus mains losses, plus a margin depending on the type of equipment, use and kind of building. The influence of the latter considerations is discussed in a later section.

Having established the margin necessary, the rated capacity of boiler or other heating unit is determined thus:

	B.T.U.'s per hour
Heat loss (e.g. insulated office block, page 33)	1,160,000
Mains losses, allow 10 per cent.	116,000
	1,276,000*
Margin (e.g. $33\frac{1}{3}$ per cent.)	425,000
	1,701,000

* This figure would in practice be checked later, after the complete system is designed against the actual emission from all piping, heating surface, etc.

A boiler, or boilers, would be selected to give as near this total as possible.

It may be assumed that the boiler radiation losses are allowed for in the margin.

To produce 1,276,000 B.T.U.'s per hour if the boiler has an efficiency of 70 per cent., and with coke of calorific value 12,000 B.T.U.'s per lb., the consumption would be:

$$\frac{1{,}276{,}000 \times 100}{12{,}000 \times 70} = \underline{152} \text{ lbs. coke per hour.}$$

4. CHOICE OF FUEL

The choice of fuel is the first decision that has to be made in considering the heating of any building.

Table XV, showing the relative cost of a therm with different fuels, gives the basic key to the question.

TABLE XV

FUEL COSTS*

Cost per therm of various fuels at 100 per cent. efficiency. (One therm = 100,000 B.T.U.'s)

	Pence
Electricity at 1d. per unit	29·2
,, ,, ,,	7·3
Gas at 9d. per therm	9·0
Gas at 4d. per therm	4·0
Fuel oil at 80s. per ton, including tax (18,000 B.T.U.'s per lb.)	2·4
Coal or coke at 60s. per ton (12,000 B.T.U.'s per lb.)	2·7
,, ,, 40s. ,, ,, ,, ,, ,,	1·8
,, ,, 30s. ,, ,, ,, ,, ,,	·9

These are exact figures not open to any argument, as they depend only on the determined calorific values of the various kinds of fuel or energy. They do not, however, without further adjustment, give the relative value of the various sources of heat represented, because they do not introduce the following factors:

(*a*) Interest and depreciation on plant.
(*b*) Heat losses in generation and transmission (i.e. efficiency).
(*c*) Convenience (including labour).
(*d*) Cleanliness.
(*e*) Facility of control.

These points are treated in detail in Chapter XVI.

The figures in Table XV and the above considerations (*a*) to (*e*) require to be borne in mind when we come to discuss the various kinds of heating systems in the next chapter.

5. TIME LAG

It will be clear that if a system were provided which would just balance the calculated heat losses at the desired internal temperature, it would in theory never be possible to attain that temperature, since in addition to the

* See note in Preface as to costs.

heat loss, the temperature of the building has to be raised. The amount of heat required for this purpose will depend on the specific heat and conductivity of the internal surfaces of the building—in other words, on the heat storage of the building.

Messrs. Grierson and Betts, in a paper read before the Institution of Electrical Engineers,* referred to an article published by one of the present authors in August 1932, in which the total weight of fifteen London buildings is given, from which they conclude that 'The heat required to raise the temperature of the structure through 1° F. is some 4·34 times the heat required per hour to maintain the temperature of the structure when the outdoor temperature is at freezing point'.

If heat losses from the building could be shut off completely, and the full heat of the installation be devoted to raising the temperature of the building from 40° F. to 60° F., it would therefore take $20 \times 4{\cdot}34 = 86{\cdot}8$ hours to do so.

With radiation and air changes, it would take far longer.

This exemplifies the enormous reservoir effect of the walls and floors of a building, which is referred to and made use of in the discussion of cooling curves and night losses (see p. 53 *et seq.*).

This also exemplifies the fact that a building first warmed in winter may take several days before conditions of comfort are reached.

A system which takes a long time to heat up or cool down is said to have a long time lag, and conversely. A building exposed to severe wintry weather may, for example, have its heating apparatus full on so as to maintain a temperature and balance all the external heat losses. A sudden change in weather conditions may occur, such as a rapid rise of external temperature. With some forms of heating it may be some hours before the heating elements can be reduced in temperature sufficiently to prevent a rise of temperature occurring in the building. The total volume of hot water and weight of metal in the boilers, mains and radiators of a heating system of the usual type may take as long as two hours to cool down (see p. 63). With panel systems the floors, walls or ceilings themselves contain a large quantity of heat which is only dissipated slowly, adding still more to the time lag. Electric radiators or gas fires can, on the other hand, be turned off immediately, either by hand or by thermostatic control.

It may be convenient to classify the time lags of the various systems in common use in the following order:

Long time lag—Panel heating with embedded pipes.
Moderate time lag—Hot-water radiators. Metal panels (water or electrically heated).
Short time lag—Gas fires. Electric radiators. Steam or vacuum steam systems. Warm-air systems.

The use of insulation inside the walls of a building reduces the heat

* *I.E.E. Journal*, May 1935.

storage and the heat losses, and therefore enables a heating apparatus of given power to catch up more rapidly with any change in conditions; i.e. the time lag of the building is shortened. In such a case it is essential, for quick response, that the heating system shall also have a short time lag.

6. COOLING TIME AND ITS EFFECT ON FUEL CONSUMPTION

From what has been said it will be seen that owing to the 'reservoir' effect of the building and heating system, grave errors will be incurred by assuming that the system will, in cold weather, be fully on during the period of occupation and closed down at other times. Operation in this way would result in the building being far below comfort-temperature for part or even the whole of the occupation-period, and in practice more heat has to be

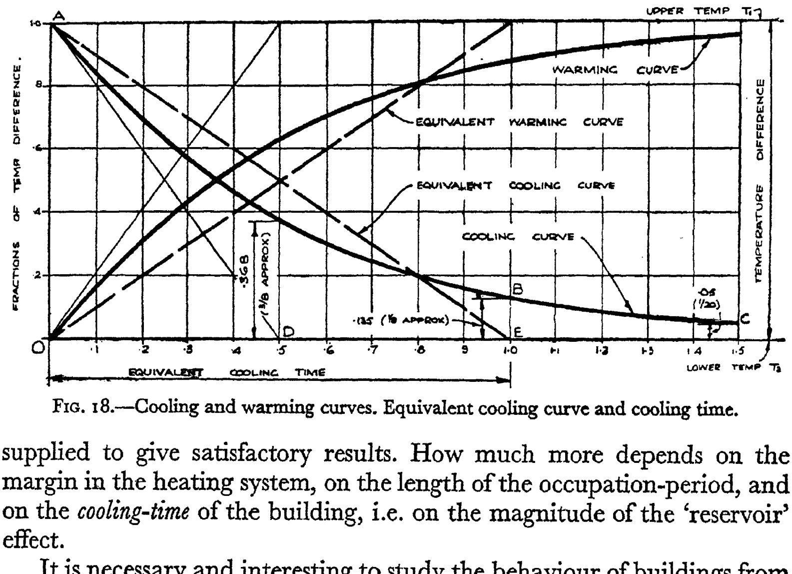


FIG. 18.—Cooling and warming curves. Equivalent cooling curve and cooling time.

supplied to give satisfactory results. How much more depends on the margin in the heating system, on the length of the occupation-period, and on the *cooling-time* of the building, i.e. on the magnitude of the 'reservoir' effect.

It is necessary and interesting to study the behaviour of buildings from this point of view, and, as a preliminary, it is proposed to consider the properties of a *cooling curve*, i.e. the curve giving the rate at which the temperature falls off when a warmed building has its heat supply cut off in cold weather.

In the case of relatively small temperature differences, such a curve is approximately governed by the fact that the rate of cooling at any time is proportional to the temperature difference.

Thus (referring to Fig. 18), if a warm object at a temperature T_1 is introduced into a place where everything is at a lower temperature T_2, the rate of cooling will at first be proportional to $T_1 - T_2$. But when the temperature falls to a lower temperature T, then the rate of cooling is proportional to $T - T_2$, and is consequently always becoming less.

It is easy to show that such a curve has the formula:

$$T - T_2 \qquad e^{-kt}$$

where k is a constant depending on the ratio of heat content to cooling surface, and t is the time elapsed since cooling commenced.

Alternatively $$\log_e \frac{T - T_2}{T_1 - T_2} = -kt.$$

Such a curve is drawn in Fig. 18. The temperature of the building never reaches the lower temperature of the surroundings (although it approaches it as nearly as may be required in a definite time). Hence it cannot be said, accurately, that there is a cooling time. Yet it would be extraordinarily useful if we could invent and define one, because we could then compare it for different buildings.

The authors have therefore introduced an *equivalent cooling curve* (shown by the line AE in Fig. 18), defined by the consideration that (1) it shall be a straight line, and (2) that the heat lost in cooling shall be the same as with the true cooling curve.

Requirement (1) means that the rate of cooling is assumed constant instead of variable, and requirement (2) that the area under ABC continued shall be the same as the area under AE.

The time OE may then be, and is defined to be, the *equivalent cooling time*.

It may be shown that the ordinates of the cooling curve at certain points have the following values:

Proportion of Equivalent Cooling-Time	Proportion of $T_1 - T_2$ (Accurately)	Proportion of $T_1 - T_2$ (Approx.)
0·5	0·368	$\frac{3}{8}$
1·0	0·135	$\frac{1}{8}$
1·5	0·05	$\frac{1}{20}$

From these facts it is easy to determine the equivalent cooling time of any building.

If the heat is suddenly shut off, we can obtain three values of the equivalent cooling time:

(*a*) Measure the time taken to drop $\frac{5}{8}$ of the temperature difference between the initial inside and the outside temperature and multiply by two.

(*b*) Measure the time taken to drop $\frac{7}{8}$ of the temperature difference, which is the actual time.

(*c*) Measure the time taken to drop $\frac{19}{20}$ of the temperature difference and multiply by two-thirds.

Conversely, to draw the actual cooling curve for a given cooling time, set up these ordinates, at times equal to one-half, one whole, and one and one-half of the equivalent cooling time and draw an even curve through the four points so obtained.

When a building has the heat turned on at a uniform rate, which will ultimately raise it from T_2 to T_1, the *warming curve* is the inverse of the cooling curve between the same limits of temperature, and the same construction may be adopted, the equivalent cooling time and the equivalent warming time being the same (see Fig. 18).

Another interesting point about the cooling curve is that the tangent of the curve at the initial temperature strikes the base line at the point corresponding to half the equivalent cooling time, i.e. AD is tangent to the cooling curve at A and OD is half OE.

This is useful both for drawing the cooling curve and for determining the equivalent cooling time.

Before we can apply this to actual buildings, as affecting night losses, there is another matter which needs study and appreciation.

In multi-storey buildings with brick walls of 14 in. or so in thickness, and the usual proportion of window, it can be shown that the reservoir effect is enormous, and that the walls gradually take up a temperature which is the mean between the average inside temperature and the average outside temperature for the season and climate.

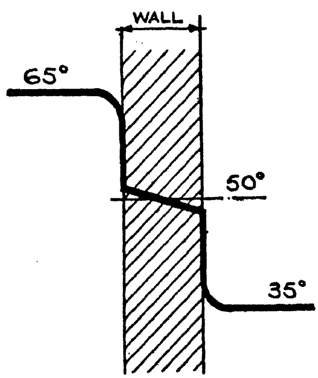


FIG. 19.

This reservoir effect is so great that actual variations of external or internal temperature affect the mean wall temperature so slowly as to be negligible over a period of 24 hours or so.

Thus, in a typical office building in London in coldest winter weather, the internal temperature may be say 65° average, and the external temperature may average 30° to 40° over a week, say 35° average.

Then the *mean wall temperature* will be about the mean between 65 and 35, *say* 50° *F.*, and the actual conditions approximate to those shown in Fig. 19.

As there is a heat transfer across the wall there will be a temperature gradient through the wall, but this is generally small and will for this purpose be neglected.

If the internal heat is suddenly cut off, an immediate falling off of temperature occurs according to a cooling curve, and the internal temperature will gradually drop till it takes up the wall temperature.

In other words, the cooling curve will be from 65° upper limit, becoming asymptotic, not to the external temperature (35°) but to the wall temperature (50°).

It is true that if the heat were left off for weeks or even days, the wall or fabric temperature itself would gradually drop, and the room temperature with it, but this effect is so slow that it can be neglected in considering night losses, or temperature logs over 24 hours.

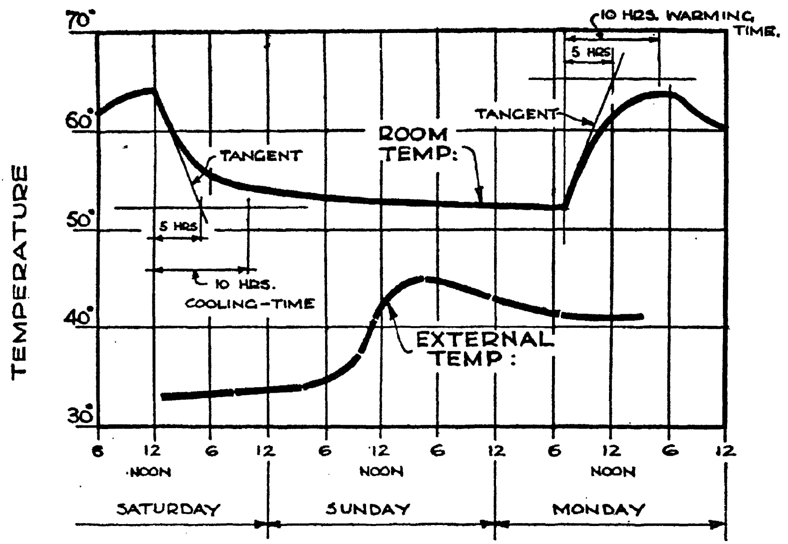


FIG. 20.—Cooling and warming curves taken in authors' office during winter week-end—heating off.

This is illustrated in Fig. 20, which gives the actual temperature log in the authors' office on a cold week-end.

The air temperature at noon Saturday was 64°, and the heat was then entirely shut off. The cooling curve is given. It will be seen that:

(*a*) The temperature log is similar to the theoretical cooling and warming curves, and becomes asymptotic, not to the external temperature but the wall temperature (about 52°).

(*b*) It is practically unaffected by the change in external temperature, showing how great the reservoir effect of the walls is.

(*c*) The cooling time is about 10 hours.

(*d*) Between 8 a.m. Sunday to 8 a.m. Monday, the temperature only dropped from 53° to 52°, though the external temperature was about 40°, again proving how slowly the walls change their temperature.

(*e*) The warming curve on Monday also gives a warming time of about 10 hours, showing there is not much margin in the heating installation.

We can now apply this to some typical examples.

The five examples in Figs. 21 to 25 all have this in common, that the total heat supplied during the 24 hours is 15 units, one unit being defined for this purpose as the heat which exactly balances the losses for one hour

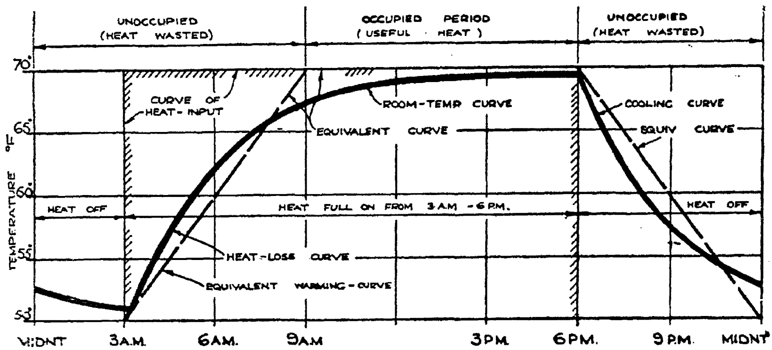


FIG. 21.—24 hours heating log, for building having an equivalent cooling time of 6 hours, and a heating system with no margin. Heat full on from 3 a.m.-6 p.m. Average wall temperature assumed 50° (mean between 70° and 30° F.).

when the building is at the desired temperature (70° in this case). In every case, the period of occupation is also assumed to be 9 a.m. to 6 p.m., as frequently applies for offices, and the external temperature 30°. It goes without saying that the heat lost in every 24 hours is also 15 units, and in

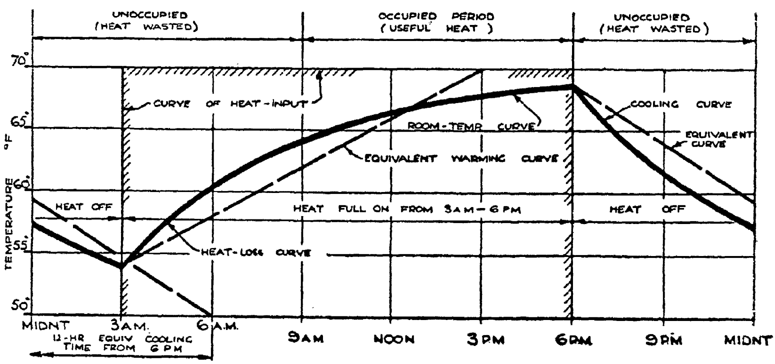


FIG. 22.—Heating log, corresponding to Fig. 21, but for a building having an equivalent cooling time of 12 hours.

each case it will be found that the area of diagrams under the cooling and warming curves is exactly the same as that in the heat input diagrams.

In Figs. 21 and 22 these 15 heat units are all supplied at a uniform rate in the 15 hours from 3 a.m. to 6 p.m., this rate being that which would just balance the losses at 70° if left on long enough, no margin in the heating system being available.

In Fig. 21 the equivalent cooling time is 6 hours, and in Figs. 22 to 25 inclusive, 12 hours.

In Fig. 23 the heat is left on all night at one-quarter full on, and full on from 6 a.m. to 6 p.m.

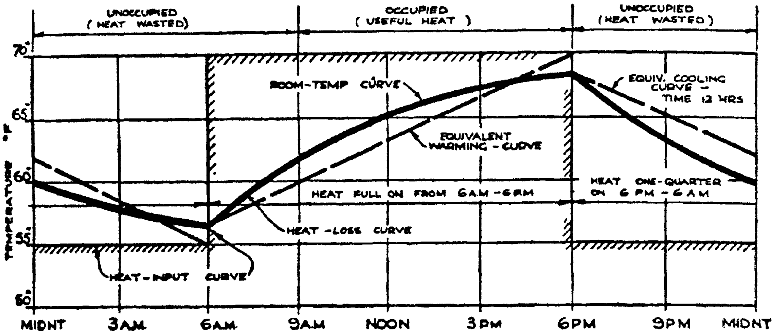


Fig. 23.—24 hours heating log, for a building having an equivalent cooling time of 12 hours, and a heating system with no margin. Heat full on, 6 a.m.-6 p.m., and one quarter full on from 6 p.m.-6 a.m. Average wall temperature 50° F.

In Fig. 24 the heating system has a 'margin' of 50 per cent. over its normal duty, and this is fully used from 6 a.m. to noon (i.e. half the equivalent cooling time).

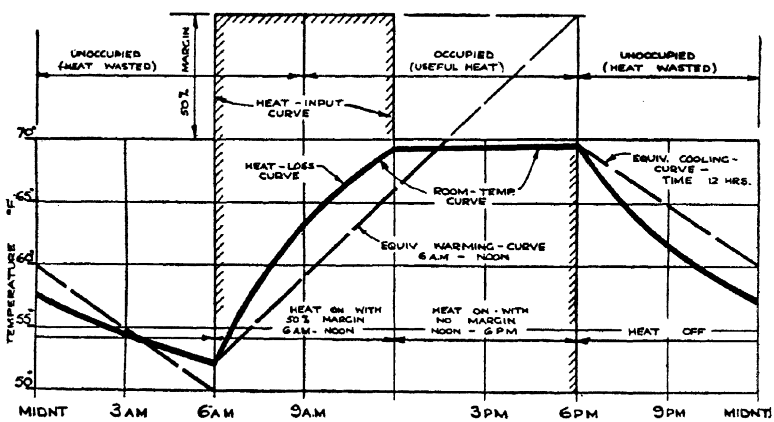


Fig. 24.—Heating log corresponding to Fig. 23, but with heating system having 50 per cent. margin, fully used from 6 a.m. to noon. No heat at night (6 p.m.- 6 a.m.)

In Fig. 25 the system has a margin of 100 per cent., which is fully used from 6 a.m. to 9 a.m. (i.e. one-quarter the equivalent cooling time).

The temperature during the occupied period may be considered as satisfactory in Fig. 21 and Fig. 25, and fairly so in Fig. 24.

The temperatures in Figs. 22 and 23 are, however, not so satisfactory, and three additional heating units (as previously defined) would be needed in further preheating to make them so.

From these and other data, the diagrams, Figs. 27 to 29, have been constructed showing approximately the equivalent number of hours full heating (i.e. the number of 'units' defined above) needed to give satisfactory conditions for various values of

cooling time;
hours occupied;
margin in heating installation.

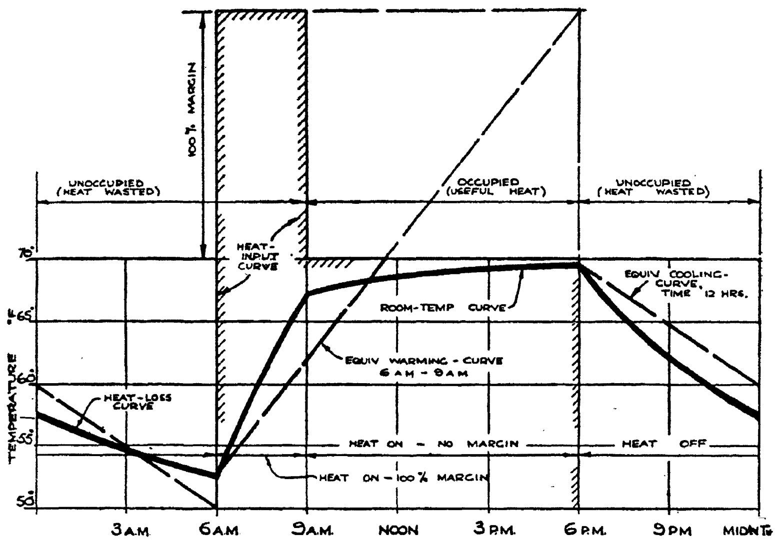


FIG. 25.—Heating log corresponding to Fig. 23, but with a system having 100 per cent. margin, fully used from 6 a.m.-9 a.m. No heat at night (6 p.m.-6 a.m.).

It will be seen that the greater the cooling time, the more is the heat required, especially when the margin in the installation is low.

By plotting the points corresponding to Figs. 21 to 26 on these diagrams the adequacy or otherwise of the systems may be determined.

The heat required naturally increases with the period of occupation. For example, it will be seen that when the occupied period is 12 hours, the heat loss is 50 per cent. of the full 24-hour loss when the cooling time is zero, but increases with a cooling time of 24 hours to $82\frac{1}{2}$ per cent. with 100 per cent. margin; $87\frac{1}{2}$ per cent. with 50 per cent. margin; and 100 per cent. with no margin.

Cooling times of six hours are common when applied to single rooms in a cold house, but when the whole house is heated, a 24-hour time is more

usual. In the former case heat is lost through all four walls, floor and ceiling, in the latter through the outside surfaces only.

It may be asked how it is that systems with little margin manage at all.

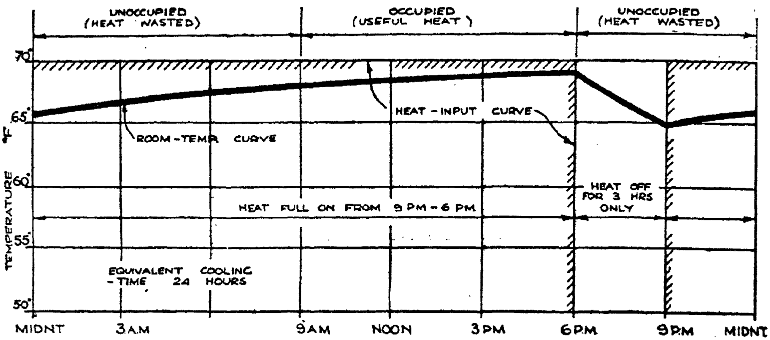


FIG. 26.—Heating log for a building having an equivalent cooling time of 24 hours, and a heating system with no margin. Heat off for 3 hours only (6 p.m.-9 p.m.). Average wall temperature, 50° F.

The answer is that in the coldest weather they need to be run practically the whole 24 hours, as shown by Fig. 26.

This is not so economical as when a large margin is available.

In more moderate weather, say, outside temperature about 50°, the

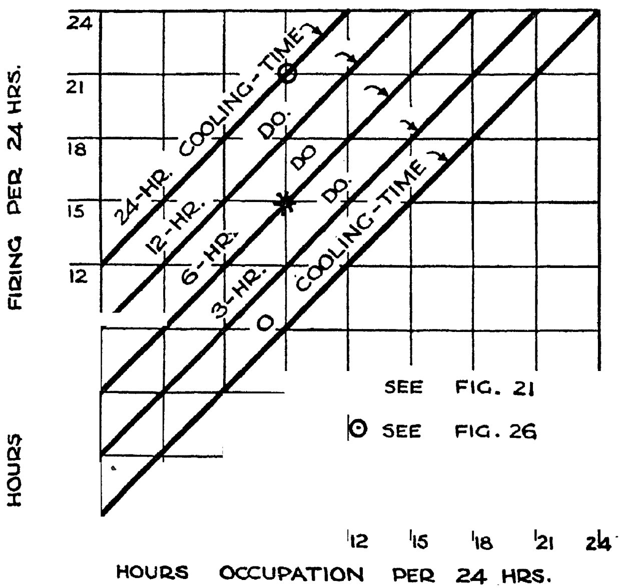


FIG. 27.—Relation between occupation-period and time of firing for installations having no margin.

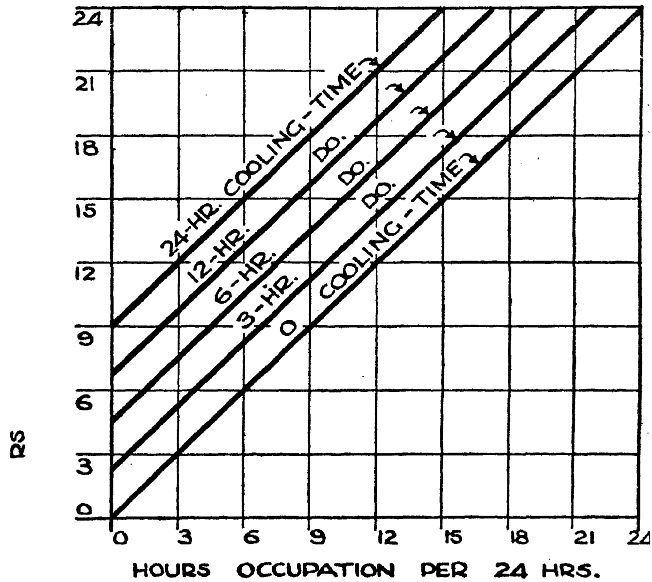


FIG. 28.—Relation between occupation-period and time of firing for installations having 50 per cent. margin.

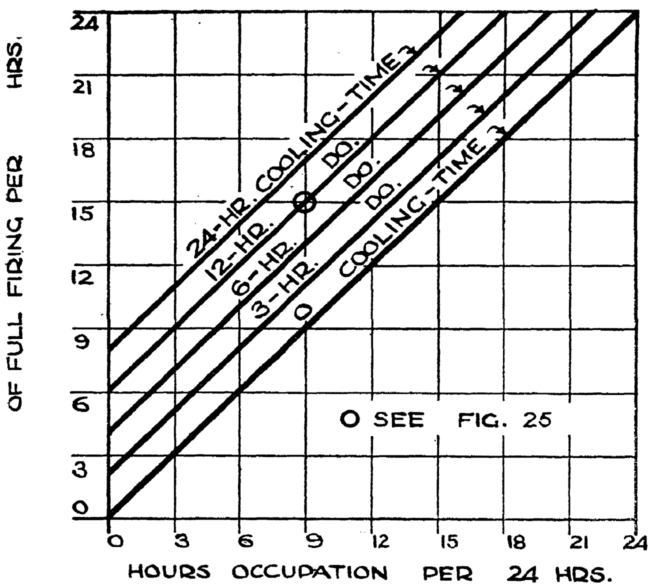


FIG. 29.—Relation between occupation-period and time of firing for installations having 100 per cent. margin.

system with no margin at 30° now has a 100 per cent. margin, and at 43⅓° F. has a margin of 50 per cent., which explains why a system with no margin at 30° works quite economically at more moderate temperatures.

But the provision of a 50 per cent. margin, even on the 30° conditions, makes for general economy and convenience.

In all the logs given (Figs. 21-26) it has been assumed that there is no time lag in the heating system itself, i.e. that it can be turned fully on or off without delay. This is almost true of direct electric heating, but the curves need rounding a little at the highest and lowest points before they can apply to oil firing, and still more to coal- or coke-fired boilers.

The system for this purpose may be classified thus:

(1) Gas fires, electric radiators, and similar systems of direct heating, may clearly be shut off completely at night until the morning warming is needed.

(2) Oil or other automatically fired boilers may be shut down completely at night, but the heat contents of the boilers and distributing system are lost during the night and have to be compensated in the morning by starting up earlier before the warmth is required.

(3) Coal- or coke-fired boilers may be let out each evening and relighted in the morning—an inconvenient arrangement—in which case the above equally applies. They are, however, more generally banked up for the night, in which case the losses incurred are those already described, and in addition the fuel consumed during the night.

This fuel is not all an extra loss, as it clearly leaves a slightly warmer condition for the start in the morning, a condition which makes for greater comfort. The fuel consumed at night may nevertheless be considerable and must be allowed for.

To give an example of the loss under (2), suppose we consider a hot-water system in the insulated office building previously referred to on p. 33, which consumes about one million B.T.U.'s per hour in 30° F. conditions.

This might have a water content in the heating system as follows:

	Gallons
Radiators of panels - - - -	600
Mains - - - - - - -	440
Boilers - - - - - - -	400

If this is at 160° F. hot, and 60° F. cold, it cools at night by about 100°, corresponding to 1,440,000 B.T.U.'s.

Added to this is the heat content of the iron in radiators, mains and boilers.

For the installation in question, the weight of metal would be:

Radiators - - - - -	60,000 lbs.
Mains - - - - - -	22,000 „
Boiler - - - - - -	14,000 „
	96,000 lbs.

multiplying this by the specific heat (0·12) and the temperature rise (100° F.) the B.T.U.'s required are:

$$96{,}000 \times 0{\cdot}12 \times 100 = 1{,}150{,}000.$$

The total heat content is then:

Water - - - - - -	1,440,000
Iron - - - - - -	1,150,000
Total - - - -	2,590,000 B.T.U.'s

The full output of the boilers being 1,701,000 B.T.U.'s per hour, it will take:

$$\frac{2{,}590{,}000}{1{,}701{,}000} = 1\tfrac{1}{2} \text{ hours (approx.)}$$

to heat the system up, even if no radiation occurred during this period.

With radiation it will take at least two hours.

With coal or coke, a further period must be allowed for getting the fire from banked condition to maximum output.

This time of two hours or more has the effect of introducing transition curves on the temperature logs and producing delayed action which involves the need for lighting up the boiler an hour or so earlier than the theoretical warming and cooling curves indicate.

The daytime B.T.U.'s being 1,276,000 for say eight hours = 10,208,000, it will be seen that the 2,590,000 B.T.U.'s needed to warm up, represents a night loss of 25 per cent.

It was shown on p. 51 that, under the conditions defined, our office building consumed 152 lbs. of coal an hour when the heating system is emitting at $1\frac{1}{4}$ million B.T.U.'s per hour. At night this system would probably consume 15 per cent. of this = 23 lbs. an hour.

So we have the following log (assuming a $33\frac{1}{3}$ per cent. margin and equivalent cooling time of 12 hours):

	Hours	Consumption per Hour	Fuel Used. lbs.
Night:			
6 p.m. to 6 a.m.	12	23	276
Morning:			
6 a.m. to 9 a.m.		203	609
Day:			
9 a.m. to 6 p.m.		152	1368
			2253

So that the additional night loss owing to fuel burnt when the boilers are banked amounts to:

$$\frac{276 \times 100}{2253} = 12\tfrac{1}{4} \text{ per cent.}$$

It may fairly be argued that some of this fuel is useful in leaving the building and water warmer at 8 a.m. next morning than would otherwise be the case, but in fact it is mostly lost.

7. TIME LAG AND MARGIN IN SYSTEM

It will be appreciated from the above that in any practical heating system it is essential to provide a suitable margin over and above the calculated heat requirements. The margin will control the time taken to attain a suitable temperature. If no margin, the time taken will be equal to the cooling time; if 50 per cent. margin, about half the cooling time; if 100 per cent. margin, about one-quarter of the cooling time.

Now in the case of *direct* systems such as electric and gas radiators, panels, etc., this margin must be represented by an addition to the calculated emission necessary in the room. A common figure is 20 to 25 per cent., though $33\frac{1}{3}$ per cent. or 50 per cent. obviously gives more responsive operation.

In the case of *indirect* systems served from a central source, such as a hot-water radiator system operated by a boiler, it is not usual to allow a margin on the heating surface installed in the rooms to be heated, at any rate nothing more than is determined by the choice of the next size radiator above that called for, giving perhaps 5 per cent. to 10 per cent. surplus at the most. It is to the boiler that the margin is added.

It may well be asked what is the use of increasing the boiler power if the whole system of pipes, radiators, etc., has not a similar margin. The answer is that during the warming up period the radiating surface will be able to emit more heat per sq. foot due to the higher temperature of water which can be achieved if the boiler margin is available. In addition, at the commencement the air temperature is perhaps only 50° instead of the normal 65 or 70, which again allows of increased emission provided the boiler power is available.

For instance, if a mean water temperature of 200° can be attained with air at 50°, the difference is 150° during the warming up period. Under normal conditions the difference would be 160° – 70° = 90. This means that there is about 66 per cent. margin obtainable on the temperature difference, which in terms of emission is about 80 per cent.

A reasonable margin of boiler power to cover these conditions appears to be about 33 to 50 per cent., depending on the cooling time. This has the additional advantage that normally the boiler will not be running 'all out', thus adding greatly to the convenience and efficiency of its working.

The margin will also allow for the difference between average conditions during normal running and the exceptional output under optimum conditions which forms the basis of makers' ratings.

Somewhat different considerations apply in the case of steam-heated surfaces in which there is little flexibility in the temperature of the heating medium.

It should be noted that as the cooling and heating curves are dependent on the construction of the building, so is the margin necessary on the heating system. Buildings of heavy construction require a larger margin than those of light construction.

8. TIME LAG WITH INTERMITTENT HEATING

What has been said above applies to buildings heated every day of the week in which the heat is let down only at nights and week-ends.

In the case of buildings such as churches, which are heated perhaps only one day out of seven, it will be obvious that the margin necessary for overcoming time lag is much greater. The fabric of the building after five or six days will have itself proceeded along its own cooling curve, so that the heating system will have to restore this heat, in addition to that of the air, and the normal heat losses.

For this reason such buildings should have an installed emission surface of the order of 50 per cent. above the calculated normal, and a boiler power with about 100 per cent. margin. Even with such provision it will be found in cold weather that the boilers have to be lighted at least a whole day and kept banked at night before the building is to be used.

If such a building is heated by direct electrical heaters, similar considerations apply. A margin of 100 per cent. would not appear to be too much for this case.

The table overleaf is quoted from the *I.H.V.E. Guide*:

TABLE XVI

Allowances for Intermittent Heating

Addition to be made to heat-loss calculation, expressed as a percentage of the total heat-loss from the room, i.e. structure and air change losses.

Buildings of Heavy Construction.

Type of Heating Apparatus	Preheating Period	Period of Occupation		
		7 days per week	5½ days per week	1 day per week
Gas or electric fires, radiators or convectors	3 hr.	40%	55%	Not recommended
	6 hr.	25%	40%	150%
	24 hr.	Nil	20%	90%
Unit heaters, or Plenum systems with metal ducts in factories	3 hr.	45%	60%	Not recommended
	6 hr.	30%	45%	,, ,,
	24 hr.	Nil	20%	90%
Low-pressure steam radiators or coils	3 hr.	35%	50%	Not recommended
	6 hr.	20%	35%	,, ,,
	24 hr.	Nil	15%	75%
		Boiler banked during periods of non-occupancy		
Low-pressure hot-water radiators or coils, with boiler having 25% margin over installed load	3 hr.	15%	25%	Not recommended
	6 hr.	10%	15%	,, ,,
	24 hr.	Nil	10%	60%

Buildings of Light Construction.

Where the thermal capacity is low, as in some glazed or sheeted constructions, these percentages may be reduced by 50 per cent. for a top floor and by 25 per cent. for other floors in multi-storey buildings.

Note: It should be noted that occasionally the temperature at 7 a.m. may be 20° F. lower than at 7 p.m. on the previous evening. It may not be possible to comply with the Factory Act (which demands 60° F. as the compulsory minimum after the first hour where a substantial proportion of the work is done sitting) if a system is not designed for intermittent heating in severe weather.

CHAPTER IV

Various Heating Systems Described

The number of different heating systems is almost unlimited, if every combination of fuel, method of firing, transmission medium, and type of radiating or convecting element is considered. It is therefore useless to attempt to describe them at all clearly or systematically unless they are classified. Though the authors have not seen this attempted previously, it has been done in many other sciences and they have ventured to propose a classification, which is indicated in Table XVII (pp. 68 and 69).

As mentioned in Chapter III, all heating systems may be divided into two main groups:

(*a*) '*Direct*' systems, in which the fuel or energy purchased is consumed in the room to be heated.
(*b*) '*Indirect*' systems, in which the energy is consumed at some more or less centralized point outside the room to be heated.

The *direct systems* are separated into four main subdivisions, according to whether *the fuel* (*or energy source*) is solid, liquid, gaseous, or electrical. Each of these is again subdivided according to the nature of the emitting or radiating element. This group is simplified by the fact that there is no transmitting medium.

The *indirect systems* are subdivided according to the kind of *transmitting medium*, which may be solid, liquid, vapour, or gaseous, and these in turn are subdivided according to their emitting element and other characteristics.

Within this great group, the fuels may be interchanged in any combination of transmitting medium and emitter, and a place is found for them in the table consistently with this.

We may comment shortly on the different systems as follows:

DIRECT SYSTEMS—SOLID FUEL

Open Fires—Open fires were, of course, the primitive source of heat, greater economy being obtained in mediaeval times by having the hearth in the centre of the room and allowing the gases to escape through a hole in the roof. Little heat was lost, but the system was not without its disadvantages.

Later, flues were invented, whereby economy was sacrificed for the greater advantage of smoke elimination.

Some of the cheap Victorian fireplaces would appear to have been designed by coal merchants, for the heated air from them went straight up

TABLE XVII

Classification of Heating Systems

	Fuel or Source of Energy				Example
DIRECT SYSTEMS	Solid		Open fires		Coal
					Coke
					Lignite
					Wood
					Peat
			Stoves		Coal
					Coke
					Anthracite
	Liquid		Oil stoves		
			Oil-burning radiators		
	Gas	Primarily Radiant	Gas fires		
			High-Temperature Gas Panels		
			Low-Temperature Gas Panels		
		Primarily Convected	Gas Radiators with water		
			Gas Radiators without water		
			Gas Convector		
			Gas Unit-heater		
	Electricity	Primarily Radiant	Luminous Radiator	Bulb Type (Fig. 30 (*a*) and (*b*))	
				Exposed Resistance (Fig. 31 (*a*), (*b*), (*c*))	
			High-Temperature Panel (Fig. 32)		
			Low-Temperature Panel	Metal Plate (Fig. 33)	
				Surface Resistance Wires	
		Primarily Convected	Low-Temperature Tubes (Fig. 34)		
			Electric Convector (Fig. 35)		
			Electric Vapour-Radiator		
			Electric Water-Radiator		

Method of Firing or Applying	Fuel or Source of Energy		Transmission Medium	Example
Magazine or Continuously-fed Boiler Automatic Stoker Hand-Firing	Solid	INDIRECT SYSTEMS	Solid	Not yet in Common Use
Oil-fired Boilers: High-Pressure Oil-Jet System Compressed-Air oil-firing Low-Pressure-Air oil-firing	Liquid		Liquid	Hot Water Low Pressure in: Exposed Pipes Radiators (Fig. 36) Flat Metal Panels Embedded Pipes (Panel System) (Fig. 37) Convectors (Fig. 38) Unit Heaters Hot Water High Pressure in Pipes, Convectors, Unit Heaters (Fig. 39) Other Liquids not yet Commercially Developed
Gas Boilers: Luminous Burners Non-Luminous Burners Natural Draught Forced Draught	Gas		Vapour	Vacuum Steam: Pipes; Radiators (Fig. 191); Metal Panels; Convectors 'Vapour' and Low-Pressure Steam: Pipes; Radiators (Figs. 186-8); Convectors High-Pressure Steam in Pipes (Fig. 40) Unit Heaters with Steam (at High or Low Pressure or vacuum) (Fig. 41, *a*, *b* and *c*) Steam Calorifiers heating water for any of the Systems (above) using Hot Water
Immersion Heaters; Electrode Boilers: Thermal Storage; Non-Storage	Electricity		Gases	Flue-Gases in Ducts—Roman System Warm Air in Ducts not Ventilating (Fig. 42) Warm Air System Ventilating: With Ducts; Without Ducts ('Pipeless' System) (Fig. 43) Plenum System (Fig. 44) (with or without recirculation) heated by: Steam; Water; High-Pressure Water; Electricity or Gas Ventilation Systems at Atmospheric Pressure with Controlled Inlets and Outlets

the flue, and most of the radiation was masked by ironmongery. Efficiency would be from 10 per cent. downwards.

Modern grates, often without bars or ironwork of any description, direct the hot gases to a sloping firebrick back which radiates out into the room, and, when burning under the best conditions, can have efficiencies of from 15 per cent. to 20 per cent. This compares with about 60 per cent. for a boiler system under average conditions. The open fire in its best form has the advantages of giving ventilation, a centre of interest in the room, and a feeling of homeliness and friendly glow.

It has the disadvantage that it burns three to four times the amount of coal required in boilers to give the same warmth, and that the coal and ashes have to be carried about, consuming labour and producing dirt, which requires more labour to remove it. The ventilation may be excessive (ten changes an hour not being uncommon), so as to cause draughts, and these draughts are specially objectionable as the replaced air is generally cold and untempered unless radiators or other heating supplement the open fire.

Many of the dirt and labour disadvantages are mitigated if peat, or (better still) wood, can be procured at a cost which can be afforded.

The labour associated with the laying and lighting of fires can be reduced if the recently developed open coke-burning fire is used. With this the ignition is effected by gas burners which are lit under the coke until the fire is well alight. The radiant heat from coke is probably greater than from coal, and the burning carbon monoxide from the top of the coke is pleasant to watch.

Open fires of any sort are, of course, impracticable for buildings of any considerable size, partly owing to the labour and inefficiency already referred to, and partly owing to the great waste of space and cost of the multiplicity of flues required.

Stoves—Anthracite stoves in this country, and the large coal stoves so common on the Continent, are much more efficient, labour saving, and draught reducing than the open fire, and will keep alight all night. The fire is not visible in them, however, and for this reason their use in this country is generally confined to halls and corridors. Their efficiency may be as high as 50 per cent., in which case they begin to approach the economy in fuel consumption of a boiler system, and are naturally lower in first cost than such systems. However, they still entail carrying fuel and ashes about in the habitable rooms and involve more labour (except in small houses) than a central system.

Coke stoves of the so-called slow-combustion type found usually in churches are less efficient than anthracite stoves and, owing to the intense local heating which they produce, are often the cause of bad draughts. These stoves also sometimes give rise to sulphurous smells due to sudden change of wind or down draught, and, on the whole, their use is not to be advised except where extreme economy in first cost precludes all other methods.

LIQUID FUEL

The liquid fuel commonly used is oil, burned with a wick in some type of burner. Oil stoves which discharge the fumes into the room are objectionable, and there are few conditions in which they can be tolerated as a permanent method of heating.

Oil often contains 2 per cent. of sulphur, which produces sulphurous acid on burning, bad for health and destructive of furniture.

Oil radiators, consisting of an oil burner whose flue gases circulate in a radiator before discharging into the room, have most of the objections of oil stoves, except that a little of the sulphurous acid may be condensed in the radiator. Cheapness and portability are generally the deciding factors in their use.

GAS FUEL—PRIMARILY RADIANT

Gas Fires—Gas fires have been greatly improved in recent years by increasing the radiation effect with special forms of radiant elements. Their radiant efficiency is about 50 per cent. (with an additional 10 per cent. convection) in the best examples working under good conditions. They have the disadvantage of depending on high-temperature radiation. The radiation has now, in fact, reached such a high temperature that many people find it unpleasant, and it is impossible to expose the skin to these rays for any considerable period without feeling an excessive dryness.

In this connection, Dr. Leonard Hill has advanced a theory that radiation from bright sources of heat is more comfortable than that from dull-red or dark sources. He finds that the latter produce a feeling of stuffiness in individuals who, like himself, are susceptible, by closing the nostrils. This condition is remedied by fanning, by simultaneous exposure to the rays from an incandescent electric lamp and by the presence of an intercepting screen of water vapour in the air, thus explaining the common practice of placing a bowl of water in front of a gas fire. It appears, however, that individuals vary enormously in their reaction to these 'nose-shutting' rays, and it may be said that the theory is not generally accepted.

Reverting to gas fires, it will be seen from Table XV (p. 51) that, allowing almost any reasonable efficiencies, gas heating will be much more expensive for continuous heating than coal or coke.

If, however, the gas fire is required not so much for house warming as for giving local and temporary comfort by radiation, say, for dressing or dining, it will be on for so short a time that it may effect a great economy over stove or open-fire heating. This, in fact, is the main use and virtue of the gas fire, namely, that its heat is instantly 'on tap'. For continuous heating, however, it is an expensive method, and for this reason does not find a place in the heating of large buildings.

High- and low-temperature gas panels are dealt with in Chap. XII.

GAS FUEL—PRIMARILY CONVECTED

Gas Radiators—Gas-steam radiators operate with a flame under a radiator partly filled with water which vaporizes and so warms the whole surface quickly. They usually contain a safety valve, and sometimes a thermostat which reduces the gas flow when the desired temperature is reached. They are open to the objections that the products of combustion usually escape to the room. The efficiency is, of course, 100 per cent.

These radiators are also made without water, in which case metallic conduction and convection of the flue gases serve to distribute the heat, the fumes passing up through the vertical tubes and escaping at the top into the room. The same objections, of course, apply. Both types of radiator are used chiefly in halls, shops, tea-rooms and other places where low first cost is important, and even then only when there is ample ventilation.

Gas Convectors consist of a gas-heated element contained in a metal box with an opening at the bottom through which the air enters and a grating at the top through which it escapes after being warmed. The convection currents are then confined and a higher air temperature is the result. The chief advantage is that the appearance is better than the plain gas radiator, and there is a reduced risk of burning to any person who touches the surface accidentally. They are used chiefly in halls, shops, etc.

Gas Unit Heaters—Gas-fired heaters are similar to their steam counterparts described later (p. 89), and may be convenient if gas is cheap and no steam or hot water for heating is available. The removal of fumes through a flue is normally essential. Gas-fired unit heaters are useful in factories, workshops, garages and other large spaces where noise is unimportant.

ELECTRIC HEATING—PRIMARILY RADIANT

All direct electric heating is 100 per cent. efficient, but owing to its cost, is mainly confined to supplementing other forms of heat, or to giving temporary comfort in isolated rooms.

Luminous Electric Radiators include large electric bulbs, like lighting bulbs, but generally run at a lower rating to give less light and a longer life. Lamps of this type (see Fig. 30 (*a*)) are generally rated at about 200 watts each, and may be about 12 in. long by 2 in. diameter. It therefore takes about five to obtain a kilowatt of heating, so that they are bulky for a given heating capacity, though they have the appearance of emitting a large amount of heat. A recent development is to enclose these lamps in a copper case with louvres, thus reflecting the light and heat twice (as in Fig. 30 (*b*)) and giving in conjunction with the warm tone of lacquered copper a psychological effect of great comfort.

Radiators with *exposed resistances* include the types shown in Fig. 31, in all of which the filament is red-hot, has a low time lag (about one minute) and gives off about 50 per cent. radiation and 50 per cent. convection. The capacity generally varies from 1 to 3 kw.

These heaters resemble gas fires in giving a very rapid response to a demand for heat, as by standing in the beam an immediate sensation of

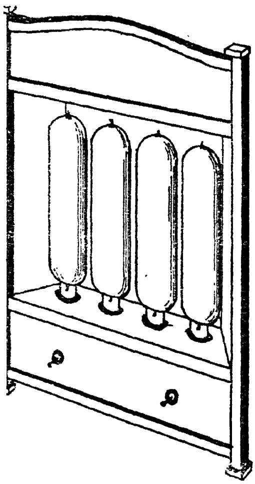


FIG. 30 (*a*)—Luminous Electric Radiator.

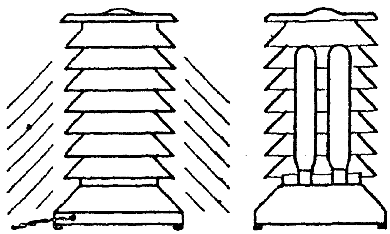


FIG. 30 (*b*)—Enclosed Luminous Electric Radiator.

warmth is felt, even in a cold room. They are therefore particularly useful in bedrooms or living-rooms used intermittently, but cannot be regarded as a satisfactory source of continuous heating, since the effect is highly localised and high temperature rays playing on the face in time become objectionable.

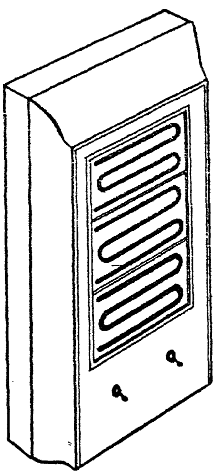


FIG. 31 (*a*).

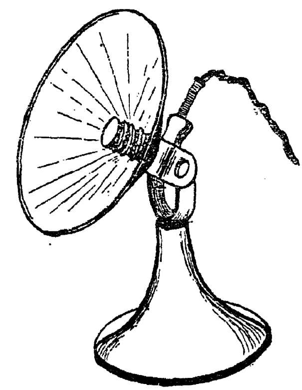


FIG. 31 (*b*).

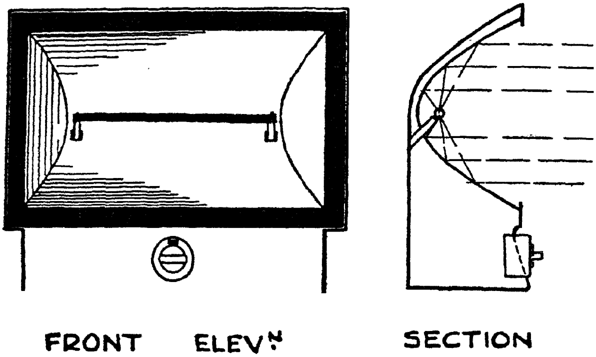


FIG. 31 (*c*).

FIG. 31 (*a*, *b* and *c*).—Exposed Resistance Radiators.

Atrocities in 'period design' are often seen; these contain a heating element and lumps of resin or yellow glass with a lamp underneath and a small rotating member throwing shadows, intended to give the idea of a coal fire. It is a moot point whether a thing which is intended to deceive has merit or demerit in proportion to the success of the deception.

High-Temperature Electric Panels, consisting of a tile of ceramic material (often about 25 in. by 13 in., 13 in. by 12 in. or 25 in. by 5 in.), enclosing a resistance element, have been used for schools and the like (see Fig. 32, p. 75). They are generally placed high up on the walls facing diagonally downwards, and reach a temperature of about 500° F. with an emission of about 2000 B.T.U.'s per sq. ft. (600 watts per sq. ft.). The radiation is about 80 per cent.

These panels are much below red heat, but would, of course, set fire to curtains, etc., as the temperature builds up when the heat loss is interfered with. They can be combined with lighting fittings.*

In this system no attempt is made to warm the air except indirectly, and thus the building may be said not to be fully warmed. The system can be so arranged that the occupants receive a well distributed heat radiation of sufficient intensity to counteract the cool air condition.

Use is made for purposes of control of a thermostat or of a device called an *Eupatheostat,* which is sensitive to radiation, air movement and sun effect, and which so regulates the electricity supplied to a heater contained in it as just to satisfy itself with radiation. When such is the case the human body, to which it conforms as regards surface temperature, is likewise stated to be in a condition of comfort.

This system is naturally of limited application but it is an interesting example of what can be done by radiation alone. It is furthermore a new departure in heating methods and one difficult of achievement with other fuels, though it now has a counterpart in gas.

* See also a paper by Mr. Hays Hallett, *Proc. I.H.V.E.*, Jan. 1931.

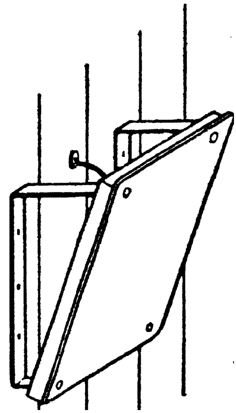
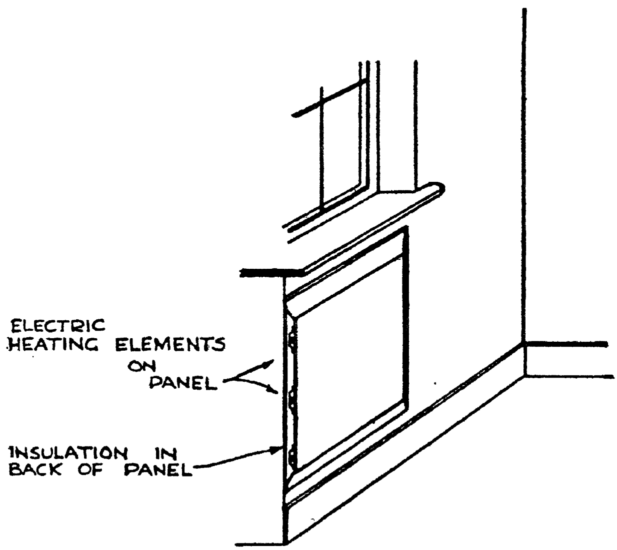


FIG. 32.—High Temperature Electric Heating Panel.

Low-Temperature Electric Panels—These may consist of a flat metal plate (as in Fig. 33), to the back of which is attached a heating element of nickel chromium wire encased in a flexible refractory material, insulated from the plate by mica strips, and covered on the back with a mattress of insulating and non-hygroscopic material. These panels generally operate at 160° F., which is the maximum temperature free from risk of burning



G. 33.—Low-temperature Electric Heating Panel.

the human skin on accidental but prolonged contact. If covered by carpet or curtain the temperature would, of course, build up.

Another system consists of electric resistance wires embedded in a kind of wallpaper strengthened by fabric, which can be stuck to walls or ceilings with heat insulating material behind. The system generally runs at about 90° F., and 18 watts per sq. ft., with about 90 per cent. radiation in still air when used on ceilings. There is no reason why it should not be as effective as hot-water panel heating, but the cost of running is usually the limiting factor.

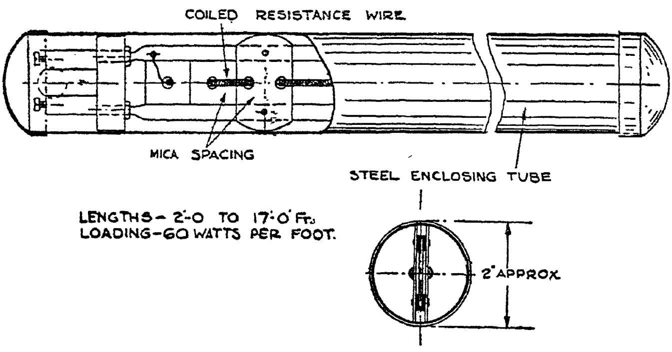


FIG. 34.—Tubular Electric Heater.

ELECTRIC HEATING—PRIMARILY CONVECTED

Heaters of this type are commonly of tubular shape, as in Fig. 34, the elements being enclosed in a thin steel tube for protection, and run at a temperature 160° to 200° F. They are frequently fixed to skirtings under windows. They have also been applied to churches, placed under the pews, in which position they tend to keep the lower air warm without heating the whole building, and this naturally leads to economy for intermittent use. Down-draughts from the upper windows are prevented by more tubes on the sills at high level.

The tubes emit about 70 per cent. convection and 30 per cent. radiation. Tubes 1½ in. to 2 in. diameter are rated at about 60 watts per foot run.

Electric Convectors generally consist of high-temperature elements in a casing with an opening below and a grating above through which an air current is induced, as in Fig. 35.

The warming effect of convectors is slower than with the radiant type of heater. A new type of portable convector containing a fan is now produced, giving a much more rapid heating effect.

Electric Vapour Radiators are ordinary cast-iron water radiators with an electric element underneath, which heats a small quantity of water

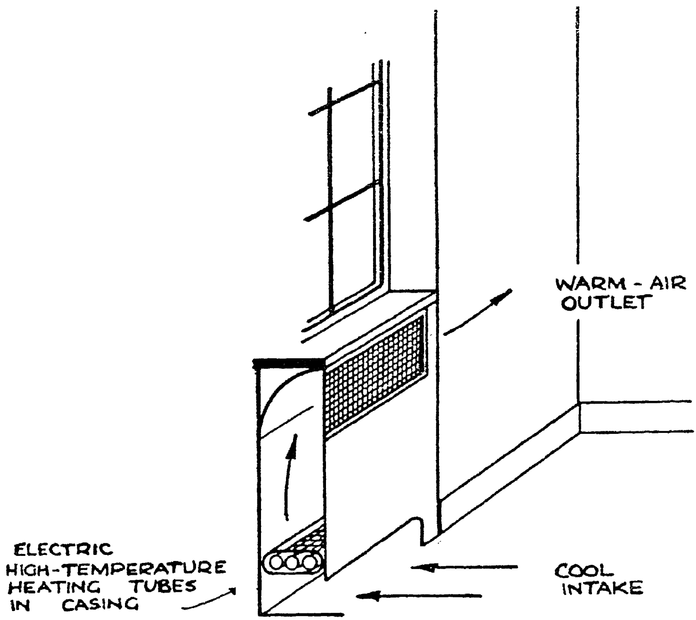


FIG. 35.—Electric Convector.

until it vaporizes and condenses on the inner surface, so heating the whole to a temperature of about 212° F. They vary from $\frac{3}{4}$ to 3 kw. in capacity.

It seems a pity, when facing the cost of direct electric heating, not to get something with a more attractive appearance, a smaller time lag, and more direct radiant effect.

Electric Water Radiators are similar but are nearly filled with water. Compared with the vapour type, the surface temperature is lower and consequently the time lag much longer, being about an hour as against twenty minutes.

Electric radiators of similar type and appearance are also made, using air inside the tubes instead of water or water-vapour.

The devising of new electric radiators has something in common with the writing of books, of which it has been well said that there is no end.

Direct Electric Heating—General—Much ingenuity has been expended by electrical engineers in attempting to prove that electrically heated buildings do not require so much heat as those heated by other means. The evidence tends to prove the contrary, and it must be admitted that with electricity at its present price direct heating is expensive. Considerable economies can be effected by having as much thermostatic control as can be afforded, but a grave warning is necessary against undersizing the heating elements in an attempt to meet the high running costs. Such a procedure is bound to lead to dissatisfaction and discomfort, besides dis-

crediting electric heating generally. Indeed, as pointed out in the last chapter under 'Time Lag', electric systems, unlike hot-water ones, can never be 'forced' even momentarily above their rated output to meet sudden demands, so that the heating load provided should be relatively more generous.

INDIRECT SYSTEMS—SOLID MEDIUM

There is at present no system in common use employing a solid medium, but heat transfer by a metallic rod from a fire (conduction) or by a moving chain passing over a fire to the place to be heated and returned (almost convection of solids) can be envisaged.

INDIRECT SYSTEMS—LIQUID MEDIA

The liquid medium employed is always water, owing to its cheapness and high specific heat.

It has the disadvantage, in some applications, that it freezes at a temperature often experienced in temperate zones, and can do much damage by its consequent expansion. Its further disadvantage, for some purposes, is that it boils at a relatively low temperature (212° F.) and produces great pressure when this temperature is exceeded.

These disadvantages are overcome in a recently invented liquid (a mixture of organic liquids) which can be heated to 480° F. without boiling, which gives much lower pressures than water at high temperatures, and which freezes without expansion at 32° F. Its disadvantage is its great cost, which has hitherto prevented its use in ordinary heating systems, though it is in use for several industrial purposes.

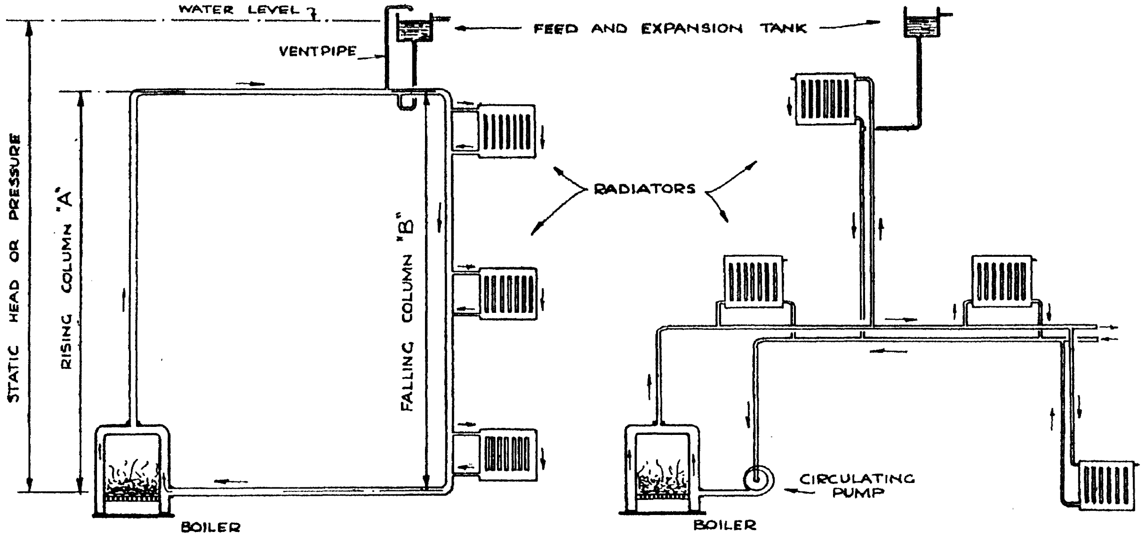
Water is still the medium implied whenever a liquid is used for heating systems.

Hot-water systems generally have the great advantage over vapour systems (including high and low pressure steam) of being in practice readily capable of regulation to meet outside-temperature changes, of being really simple, safe and silent, and of providing a surface at a moderate temperature which neither burns nor produces stuffiness by roasting the dust particles.

They have, however (continuing the comparison), the disadvantages of greater time lag, a greater surface of radiator required, a greater static pressure (which matters more in high buildings) and greater cost. Nevertheless, except in special cases (factories, garages, etc.), their greater cost is well justified by their advantages.

Exposed Pipes—This is the cheapest form of heating surface, of which the early greenhouse system is a well-known forerunner. Cast-iron and steel pipes (2 in. to 4 in. diameter) are still commonly used in factories, garages, store sheds and the like.

Hot-Water Radiators—Radiators are too well known to need detailed description. The type usually used is made of cast iron, illustrated fully in



(a) Gravity Circulation. (b) Pump Circulation.

Fig. 36.—Hot-Water Radiator Systems.

In (*a*) (gravity circulation) circulation is produced when the boiler fire is lighted, since the weight of column B is greater than that of column A, due to the higher temperature of the latter. In (*b*) (pump circulation) circulation is produced by a pump, which may be driven by electric motor or other means; this enables smaller pipes to be used and the water to be transmitted over greater distances than with gravity circulation.

various makers' catalogues. The word 'radiator' is a misnomer, since the heat is only about 20 per cent. radiant, the rest being convected.

Another type of radiator is formed of pressed-steel plates welded together round the edges to form a water space. A typical radiator system is shown in Fig. 36 (*a*) and (*b*).

Flat Panels may be either cast iron, with waterways cast on behind (see Plate XII, facing p. 177) or of wrought-iron or mild steel sheets with tubes brazed or welded thereto. They may be built into the wall or ceiling with insulating material behind them, and in this position give about 60 per cent. radiation on walls and 90 per cent. radiation on ceilings. Or they may have ribs on the back, and stand clear of the wall with space for an air current (see Fig. 103, p. 181), in which case they are intermediate in function between the ordinary 'radiator' and the true flat panel. For this type the heat by radiation falls from about 60 per cent. to 30 per cent., the convected heat, of course, gaining correspondingly.

Embedded Pipes—The 'panel' system, so called, consists of pipes embedded in the ceiling, walls or floor. The pipes are generally in copper for work of great permanence and in iron for the more commercial type of building, and are normally of ½-in. bore. When used in ceilings they are laid at about 4½ in. to 6 in. centres with the underside flush with the soffit of the concrete slab; the whole is then plastered over.

The water temperature is 130° F. or lower, and the plaster temperature at the surface varies from 100° F. at the pipes to 90° midway between them. When in ceilings, about 90 per cent. of the heat emitted is radiation.

Fig. 37 (p. 81) shows diagrammatically the arrangement of a hot-water ceiling panel and connections, and Plate III (facing p. 49) shows copper heating panel pipes for a city office building, laid on the shuttering ready for the reinforcement and concreting over.

When used on walls, the radiation from this type of panel will be reduced to 60 per cent. and to 50 per cent. when in the floor. The ceiling position is generally the most satisfactory, since furniture cannot intercept the radiation, as may happen with wall heating. The floor position is generally confined to entrance halls, etc., where people do not stay for appreciable periods with their feet in contact with the warmed surface, though with an increased area of panel at a lower temperature the arrangement has been used in schools and churches with considerable success.

When on ceilings no appreciable convection is induced, and hence a feeling of stagnation may be experienced unless some ventilation, natural or mechanical, is provided. This is in any case desirable with any system of warming buildings intended chiefly for personal occupation.

The panel system will be further discussed in Chap. VIII, but its main characteristics may be summarized here briefly as follows:

(*a*) Dirt particles are not lifted from the floor in induced air cur-

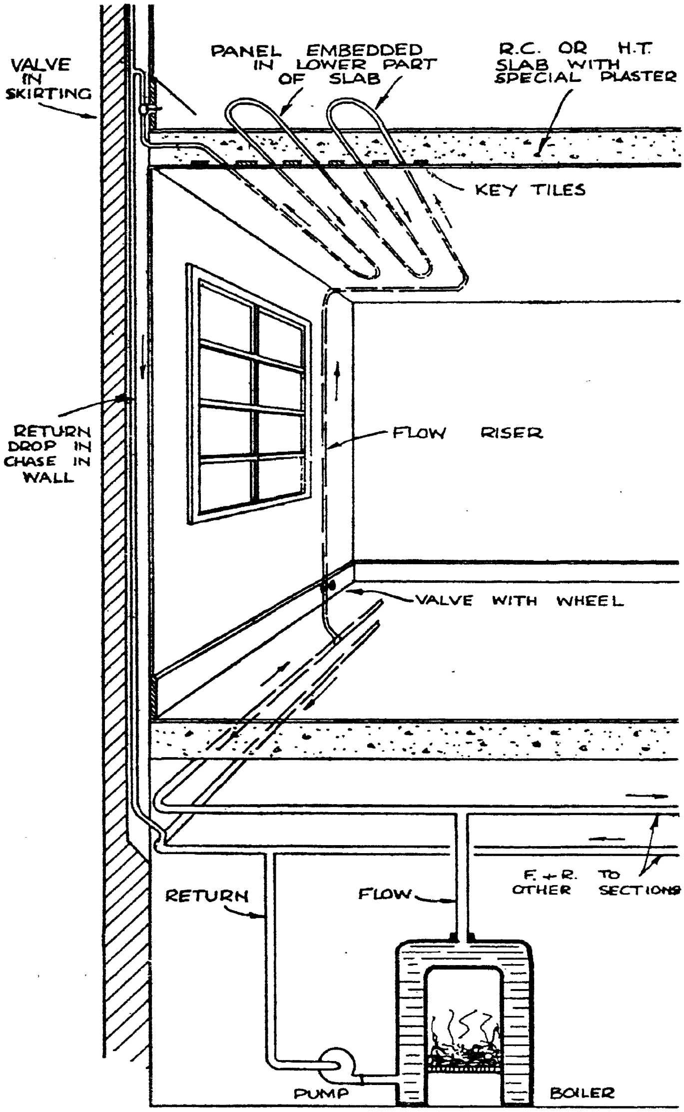


FIG. 37.—Hot-Water Embedded Panel Heating System.

rents, thus eliminating the black markings on walls and ceilings so well known with ordinary radiators, especially when run at a high temperature and placed near walls.

(*b*) In consequence of (*a*), less frequent decoration, with its cost and

interruption of occupation, is needed. As a rule, the periods between redecoration may be doubled.

(*c*) Also in consequence of (*a*), occupants do not breathe the dirt raised from the floor by convection currents.

(*d*) Very uniform warmth over the whole room.

(*e*) Complete freedom for architectural treatment and design, absence of space wasted by radiators, and elimination of dust pockets.

(*f*) The panel system is largely free from the obvious loss which occurs with a radiator system in which all the heat is, in general, supplied underneath the windows, resulting in a rising column of warm air over the face of the glass. It should be mentioned that even with the panel system the prevention of down-draughts necessitates the provision of radiation surface beneath windows where the latter are of more than ordinary height. Such surface is, however, only a small proportion of the total, and the loss is to that extent reduced.

(*g*) Panel heating has a long time lag, owing to the great weight of concrete warmed. This is a disadvantage for buildings such as halls, theatres, cinemas, shops, etc., where a sudden concentration of people may occur, but in the case of offices, flats, houses, public buildings, hospitals, and so on, the long time lag does not seem to be a drawback in practice.

(*h*) It is frequently maintained that an air temperature 2° or 3° lower than that necessary with radiators suffices with a panel system, but from what has been said in Chap. III, it will be seen that this point is a doubtful one to make with the ordinary individual. It is clear, however, that the absence of any appreciable temperature gradient in panel-heated rooms leads to lower transmission losses at the upper levels.

(*j*) Owing to the large percentage of radiation, a comfort effect is achieved even with some measure of through-draught. A radiator system, depending chiefly on convection—i.e. the warming of the air and its circulation to the individual to be warmed—clearly breaks down if the warmed air is immediately blown out of the nearest window. Panel heating is, therefore, particularly suitable for schools, hospital wards and other places where thorough ventilation is desirable.

It follows from (*f*), (*h*) and (*j*) above that a reduction of fuel consumption is associated with panel heating under such circumstances, and this is borne out by experience.

Hot-Water Convectors—A hot-water convector generally consists of a

finned air heater so placed as to induce an air current, as shown in Fig. 38. It is a neat and compact form of heating, but being entirely convective suffers from the disadvantages associated with the ordinary radiator as regards the carrying of dust and dirt by the air from the floor to the walls and breathing zone.

In completely eliminating radiation it is contrary to the modern trend with its associated fuel economy. On the other hand, the emission is greatly increased by the induced air current and the chimney effect of the flue, and hence a lower weight of metal and water suffices for a given output. This

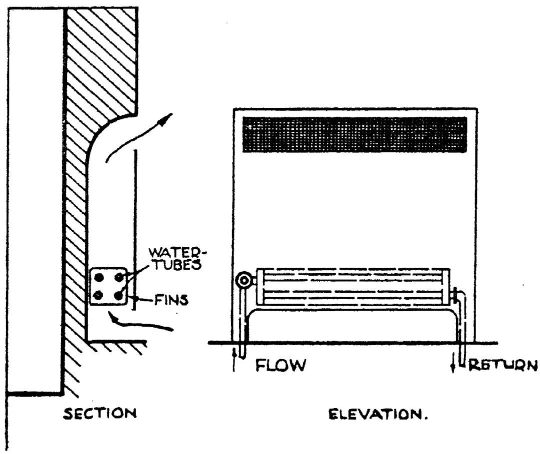


FIG. 38.—Hot-Water Convector.

gives the convector a lower time lag than an ordinary radiator, a point in its favour; and it is probably less obtrusive and more amenable to architectural treatment.

The absence of the radiant component makes it unsuitable for places requiring good through-ventilation for reasons already given under panel heating. The system is much used in France and America, especially with steam.

An interesting feature is that the emission increases rapidly with the height of the flue, being nearly doubled when this is increased from 12 to 36 inches.

Unit Heaters—These are similar to the steam types illustrated in Fig. 41 (*a*) and (*b*) (p. 89), and consist essentially of an electrically driven fan blowing cold air over a heating coil, generally of the motor-car radiator type, and thence over directing louvres back into the room to be warmed. The units are generally slung overhead and the louvres directed diagonally downwards so as to bring the warm air to breathing level. They are more frequently used with steam or high pressure hot water.

The principal is nevertheless the same with water as with steam. With

water at ordinary radiator temperature unit heaters have not in some cases been altogether successful; a temperature of water above boiling point would appear to be desirable. A large number of small units are better than a small number of large units if an adequate circulation of warmed air to all parts of the building is to be ensured. When water is used, pump circulation is essential.

Unit heaters are eminently suitable for large factories, garages and similar buildings where noise is not considered objectionable, and for such purposes are economical in first cost.

Floor type unit heaters are described later under 'steam', and are not commonly used with low-pressure hot water owing to the large output and compact size which is possible with steam.

High-Pressure Hot Water—The development of this system is of comparatively recent origin. Water may be heated to a temperature above normal boiling point if it is retained under pressure; at 150 lbs. per sq. in. gauge, for example, the boiling point is 366° F. With water in this condition circulated in a system, numerous advantages accrue for large-scale heating plants of industrial type.

As compared with low-pressure hot water smaller pipe sizes are possible, due to greater temperature drop permissible. Full advantage can be taken of compact high duty heaters of the unit type; there is also great scope in its application to process work such as drying chambers, heating of vats, etc.

As compared with steam, losses due to steam traps and re-evaporation of condensate are saved, maintenance is less and the temperature can be reduced in milder weather over a greater range. On account of these factors a high-pressure hot water system usually has a lower fuel consumption than the corresponding steam system.

The details of this system are discussed more fully in Chapter XV. A typical example is shown in Fig. 39. Water is circulated from the lower part of a steam boiler, the steam acting simply as a cushion maintaining the pressure. As the system heats up the water expands and the level in the boiler rises; as it cools this level drops again. There are numerous devices adopted to prevent steaming in the pipes, dealt with later.

An older form of high-pressure hot water system known as the 'Perkins System' requires brief reference. It is still found in use in churches, chapels, shops, halls, etc., though the number of new installations must be few on account of its numerous disadvantages such as high temperature of surface, danger of explosion, risk of burning, bad distribution of heat and low efficiency. It consists of a continuous closed circuit of strong 'Hydraulic' pipe of $\frac{7}{8}$ in. bore, jointed with metal-to-metal right- and left-hand threaded joints, containing water with an anti-freezing mixture. The pipe continues inside the furnace in the form of a coil set in brickwork. At the highest point of the system is provided a sealed expansion cylinder of strong piping, partly filled with water and partly with air. When the fire is lighted, the water circulates by gravity and the expansion is taken up in

the cylinder, thus compressing the air and raising the pressure. Temperatures of about 300° F. were usual for ordinary heating, but 500° or more have been used for baking, enamelling, drying, etc.

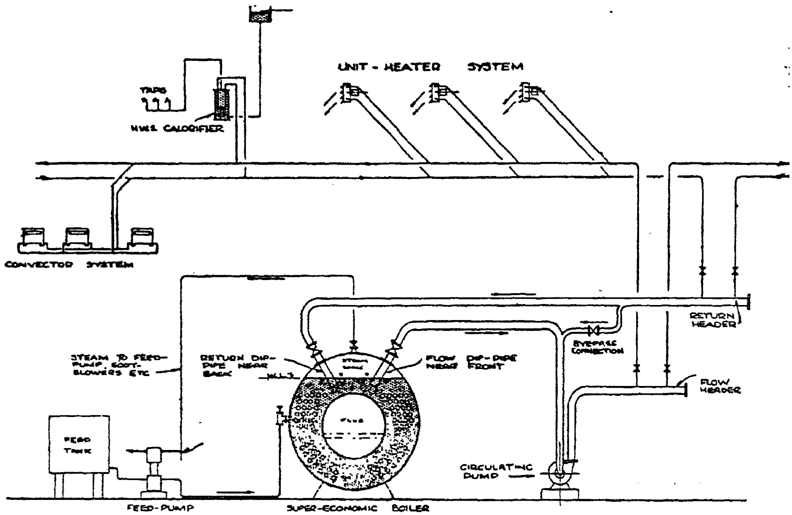


FIG. 39.—High-pressure Hot-Water System.

Hot-Water Systems Generally—From the above descriptions of hot-water systems it will be apparent that they could be classified on the basis of temperature of emitting element. Thus high-pressure hot water would come at the top of the scale and the embedded panel system at the bottom, with all the other systems between.

The great choice of systems which this range of temperature gives, together with the advantages of ease of control, simplicity and silence, no doubt accounts for the general acceptance of hot water as the medium of heating in this country for buildings of every class and description.

INDIRECT SYSTEMS—VAPOUR MEDIUM

The whole class of vapour or steam systems is subdivided in respect of pressure of operation, or, in other words, temperature of emitting element, just as with the hot-water systems referred to above.

This classification in the case of steam systems results in the following:

System	Pressure of Steam	Temperature °F.
Vacuum	25 in. to 0 in. mercury col. vacuum	133 to 212
Vapour	About 8 oz. per sq. in.	212
Low pressure	5 to 25 lbs. per sq. in.	227 to 267
Medium pressure	25 to 80 lbs. per sq. in.	267 to 324
High pressure	100 lbs. per sq. in. and over	338 and over

It will be observed that the lowest temperature (133° F.) of normal

steam operation is comparable with that at which low pressure hot-water systems are commonly worked in mild weather.

Vacuum and Sub-Atmospheric Steam Systems—This class of system (Fig. 189, p. 312) consists of a low-pressure steam boiler connected to main piping serving radiators, pipe coils, and convectors, as with a hot-water apparatus. It is usually considered inadvisable to apply this system to 'panel' systems owing to its high temperature of operation, which would be liable to cause cracking of plaster and an unpleasant intensity of radiation.

The steam condenses in the various emitting elements and returns through traps by a system of return-pipes connected to a vacuum pump. The water is returned to the boiler and the air separated out and discharged to atmosphere. There are numerous devices and complications which have been introduced into the system by various makers to improve its operation or range, so that there are several different versions in existence, though all are the same in principle.

The purpose of the vacuum is to increase the flexibility or range of temperature over which the system works, since, as the pressure is reduced below atmospheric, water can be made to boil at temperatures much below 212° F. The limit is set in practice by the commercial efficiency of vacuum pumps and by the air leaks into the system which tend to destroy the vacuum, so that temperatures much below about 133° F., corresponding to a vacuum of 25 in. of mercury column, are difficult to maintain for considerable periods.

A disadvantage which has to be guarded against by careful design is the noise of hammering which may occur, particularly on warming up. This is common to almost any steam system, and is due either to the sudden contraction of a volume of steam into a globule of condensation or to the impingement at high velocity of condensation against the walls or bends of the pipes. By careful installation this defect can be minimized, though not entirely eliminated.

The advantages of the vacuum system are:

(*a*) Short time lag.
(*b*) Leaks, if any, are inward and not outward, and hence do not cause the trouble of water leaks.
(*c*) Return pipe sizes are small.
(*d*) Radiator sizes are less than with water.

If combined with thermostatic control this system is much improved. It cannot be said, however, that it possesses the general application to a wide variety of types of building of the low pressure hot-water system.

'Vapour' System—This is similar to the vacuum system but utilizes steam at a few ounces pressure, like steam from a boiling kettle, in fact, and the condense is passed from the radiator by means of a thermostatic trap which passes water and air but is closed by steam.

The system possesses one advantage over the low pressure steam system in that by fractionally opening, or 'modulating', the radiator valve, the radiator may be partially filled with vapour, thus affording some measure of control.

The system is illustrated in Fig. 188, p. 310.

Low Pressure Steam—This system is supplied with steam from a low-pressure boiler operating above atmospheric pressure generally at about 5 lbs. per sq. in., so that the surface temperature of the pipes or radiators is about 220° or more, and the condensation is returned to the boiler by gravity and not by a vacuum pump (see Fig. 186, p. 309).

It has no flexibility, since a pressure must be maintained for the even distribution of the steam, and the heat is therefore full on or completely off. In this climate, where the average outside temperature is 40° or 45° in winter, this is an undesirable feature, though in colder countries such as the United States, Canada, and central Europe this disadvantage is not so apparent. Owing to the high temperature of its heating surface the strong convection currents and baking of the dust particles common to all intensely hot elements occurs. This produces stuffiness in the atmosphere, which is generally objectionable.

The running cost of this method is higher than hot water due to its inflexibility, though the initial cost of low pressure steam is less on account of the smaller radiators and pipes which may be used.

Medium and High Pressure Steam—Its use direct in heating apparatus is confined to industrial systems. It is used in pipe coils in factories, such as the old type of Lancashire cotton mill, as illustrated in Fig. 40. The high pressure boiler plant in such cases was often used for driving the mill engine, and the steam taken off direct for heating gave a simple reliable system. Fig. 40 shows an open return, but frequently the condense passed straight back to the boiler with a dip pipe.

H.P. steam is now more commonly used in unit heaters or plenum plants, or as a medium for the transmission of heat over long distances. In the latter case it is often reduced in pressure at the end of its journey to feed a low pressure or even a vacuum system; or again it may be passed into a calorifier for the heating of water for a low pressure hot water system.

Unit Heaters with Steam—Unit heaters (see Fig. 41 (*a*), (*b*) and (*c*)), have already been described under the heading of hot water, though they are in much greater use with steam or high pressure hot water for the reasons already given. They are a useful and economical method of space heating where the noise of the fan is not a drawback, and are easily controlled thermostatically by the simple means of shutting off and re-starting the fan. They have the added advantage, if it is needed, of promoting a brisk circulation of air in the summer, and, as is well known, this often produces a feeling of coolness due to the evaporation of moisture from the skin.

Type (*a*) is for use up to mounting heights of about 20 ft. Above this the

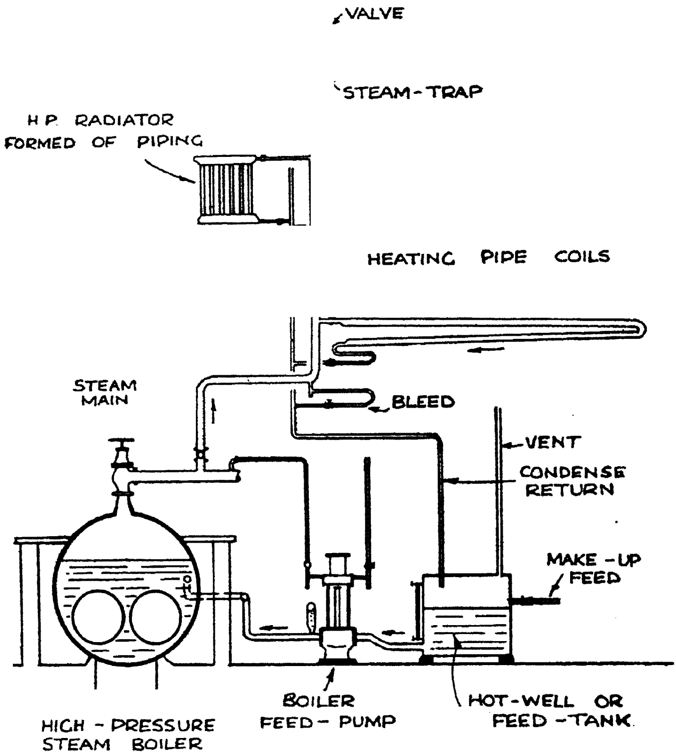


FIG. 40.—High-pressure Steam System.

down-flow type (*b*) is preferable. Type (*c*) standing on the floor is in common use, generally in the heavier types of factory or workshop. With this type, large outputs in a small space can be provided, and a type of fan incorporated capable of projecting a stream of warm air for a distance of about 120 ft.

Unit heater systems tend to suffer from the disadvantage of 'layering' of the air, often leaving the feet cold and the head warm, and to overcome this a wide variety of types has been produced claiming to eliminate this trouble by careful design of the discharge louvres, or by the proper control of the velocity and temperature of the air. On the whole it may be said that a large volume at a low temperature is better than a small volume at a high temperature, and provided this is borne in mind in the choice of the unit, the system can be most successful.

Steam-Heated Calorifiers—A calorifier is really a heat exchanger consisting of a battery of pipes fed with steam, enclosed in a cylinder containing the water to be heated.

The system is employed where steam boilers are provided for other

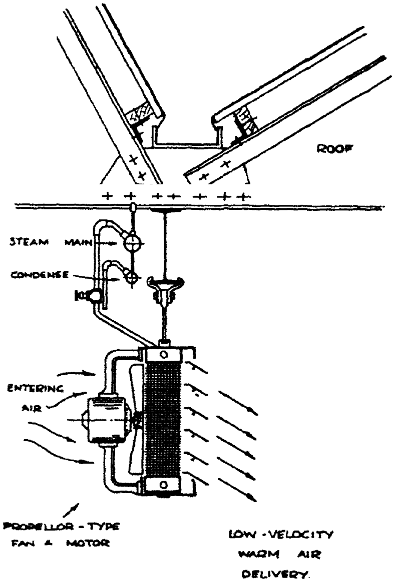


FIG. 41 (*a*).—Suspended Type Unit Heater.

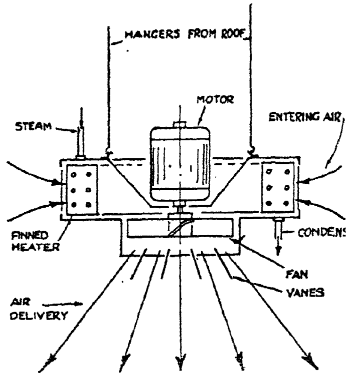


FIG. 41 (*b*).—Downward Discharge Unit Heater.

purposes and where hot water for heating the building or buildings is required from the same source.

Thus in a hospital steam is often necessary for cooking, sterilizing and laundry, yet hot-water heating and hot-water supply are required for the buildings. For these, calorifiers would be provided with only one set of boilers of steam type.

In the classification table this system should strictly therefore be placed under a heading 'Dual Media', since both vapour and liquid are employed in the transmission of heat from the source to the point of use.

Exhaust-Steam Systems — Where exhaust steam from engines or electric generators is available this may be used with advantage for heating for reasons which will be discussed in Chap. XX. Where such application is possible it may be used direct in vacuum or low pressure steam apparatus, or through calorifiers for the heating of water.

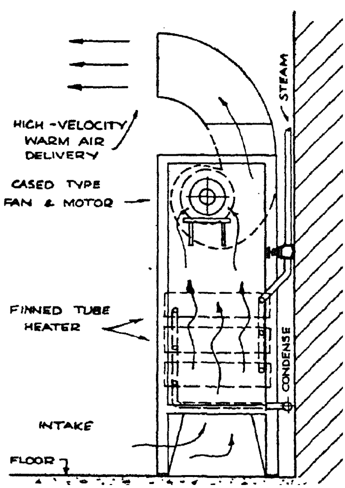


FIG. 41 (*c*).—Projector Type Unit Heater.

INDIRECT SYSTEMS—GASEOUS MEDIA

Flue Gases in Ducts (Roman System)—The culture of the Romans developed a type of heating which produced much the same effect as our radiant systems of to-day by warming the floors and walls from a furnace in the basement. Examples are brought to light from time to time in this country, and those at Bath are quite well known. Excavations at Verulamium show clearly that this form of central heating was an essential item of all the better-class houses of the time, yet the art was evidently completely lost when the Romans departed, and it has taken some sixteen hundred years or more to get back to their state of civilization in this respect.

Even to this day there lingers a prejudice against any form of warming in the home other than by the open fire, which, after all, is but a refinement of the method adopted in the Dark Ages after the Roman influence had died out.

Their system comprised a furnace below the ground from which the hot gases were conveyed in ducts under the floor to flues in the walls,

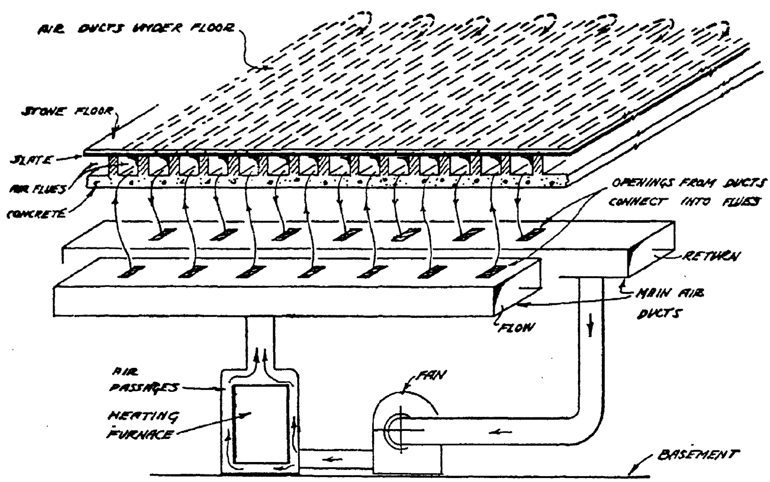


FIG. 42.—Diagram of Floor-Heating System in Liverpool Cathedral.

emerging to the atmosphere at various points around the building. Sometimes proper ducts were formed, and in other cases the whole space under the floor, known as a 'Hypocaust', appears to have been used. The wall flues were invariably of hollow tiles, very similar to their modern counterparts. The floor consisted of a slab corresponding to our concrete slabs supported on 'pilæ', and over this a mosaic was laid. The heat was furnished by wood or charcoal. Probably the distribution of heat was not uniform, but there can be no doubt that the general effect was highly successful.

Warm Air in Floor Ducts (Liverpool Cathedral)—A comparison of the Roman system with that at the new Liverpool Anglican Cathedral (Fig. 42) shows that the principle is the same, though the medium is different.

At Liverpool, air is warmed by a furnace, and circulated through the ducts beneath the floor by means of a fan. This gives much better distribution of heat and avoids the danger of the gases depositing soot in inaccessible places. The result is most satisfactory, and it is found that an average floor temperature of 68° F. is all that is necessary to maintain 60° in the Cathedral in the coldest weather, though it must be remembered that occasional cold spells will be balanced out by the enormous 'reservoir' effect of the structure. A special convecting system has been installed to counteract the down-draughts from the large windows, and it is not certain that the building would be free from draught at floor level without these.

Those responsible for the enterprise at Liverpool deserve to be congratulated on their courage in returning to so old a system and applying it in a new form. Whether this system is suitable for buildings other than those of a monumental character is open to question, as considerable difficulties are apparent if one considers applying it, for example, to a modern multi-storey building such as a block of flats or offices. Moreover the power required to circulate a given amount of heat with hot air is very much greater than with hot water, and the large fan power will be reflected in the running costs.

Warm Air with Ducts (Ventilating)—The more common method of utilizing air as the medium of transmission comes under the category of *Warm Air Ventilating*, by which is meant that the warm air is discharged into the room to be heated.

The only difference between this method and the 'Plenum' system referred to later is that in the 'warm-air' apparatus the air circulates by gravity through the ducts, due to the difference in temperature between the hot rising and cold falling columns of air; whereas in the Plenum system the air is impelled through the ducts by means of a fan.

Thus, owing to the limitations in the sizes of ducts, the warm-air system is applicable only to residences and small buildings, but the Plenum system may be used in any size of structure, even to the very largest.

A warm-air system is cheap to instal, as it consists simply of a cast-iron fire pot and flue chamber, enclosed in a steel casing connected to a series of metal ducts terminating at the inlet registers in the various rooms; the natural convection currents set up when the furnace is heated serving to convey the warm air to the points required. The ducts should be lagged to prevent heat loss, and a humidifying pan of water often forms part of the furnace. The air to be warmed is generally introduced from outside, but may be arranged for re-circulation from the building in order to save fuel. A basement is essential for the furnace.

The system suffers from the defects of any purely convection system,

but in an aggravated form, since the air may often reach 170° F. at the gratings and probably more round the furnace. These high temperatures produce the roasted 'burnt iron' smell and devitalized effect which is so difficult to account for scientifically, but which is generally thought to be due to the carbonization of the dust particles. (It is known, for instance, that dust present in the air decomposes at about 150°, and that cotton becomes carbonized at 160° F.) The distribution tends to be affected by winds, and layering of the air is difficult to avoid.

In this country the system has never found favour, though some of the older churches still use it, and it comes into the public view from time to time when a congregation or a clergyman has been overcome by fumes. These occurrences are due to the firepot having burnt out, with the result

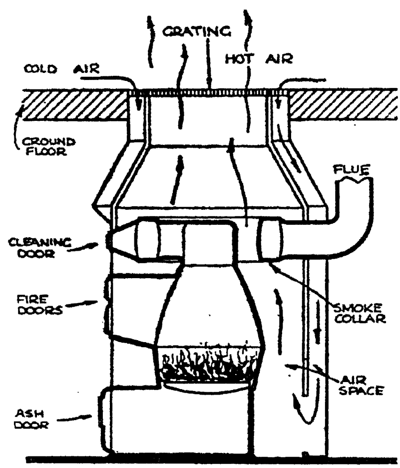


FIG. 43.—Section of 'Pipeless' Furnace for Warm-Air Heating System.

that poisonous carbon monoxide and other gases find their way direct into the building.

As with other high-temperature systems, such as the steam system, we find warm-air heating much in use in America and on the Continent. That the intricacies of design have been developed to a high state of perfection is apparent from the amount of research and literature on the subject emanating from these sources.

'One-pipe' or 'Pipeless' Heaters—These are really identical in principle with the warm-air system as described above, except that they have no distributing ducts (see Fig. 43). The furnace, enclosed in a casing as before, is placed in the basement immediately under the hall, where it discharges the warmed air through a grating in the floor. The heated air rises naturally up the stairwell to the top floor and enters the various rooms through any doors which have been left open. The cooler air which is displaced falls in the same way and returns to the heater through the outer

portion of the same grating in the floor of the hall, this portion being divided from the centre.

The system is known to work quite well in cases such as that described above, but it has failed when attempts have been made to apply it to long rambling houses where some definite method of distribution is essential. It can never be said to warm the whole of a house, but in heating the hall and central portions it does perform a very useful function.

The Pipeless or One-pipe system has the merit of cheapness in first cost and simplicity in operation. It cannot be looked on as a complete system of heating, nor is it applicable to more than a small proportion of the various types of houses that are built, whilst for other and larger buildings it is entirely unsuitable.

Plenum System—The Plenum system of heating comprises a centrifugal fan; a heater warmed by steam, hot water, electricity or gas, and a system of ducts distributing the air to the points required, as in Fig. 44.

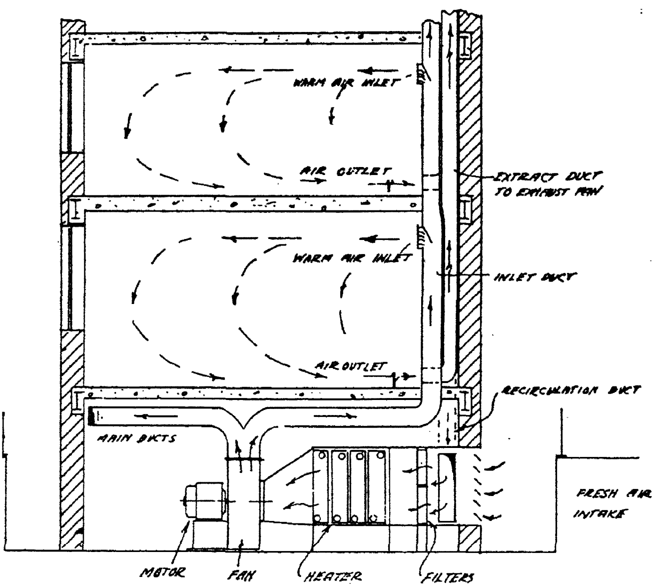


FIG. 44.—Plenum Heating System.

Air is drawn in at the fresh air intake and is warmed at the heater to a temperature generally between 90° and 120° F. When this air is discharged into the room to be heated it cools to the room temperature, so giving up its heat to the walls, floor, ceiling, etc., just as the heated air from a convection radiator system cools down in its circulation through a room. There is, however, an important difference between these two systems in that the warmed air of the Plenum system is under a slight pressure, so that

inward leakages of air through windows, etc., usual with radiator systems, are non-existent, and all air leakages are outward.

This air leakage means that a great loss of heat is continually taking place, since all air supplied by the fan has been warmed in winter from, say, 30° to 120°, and has cooled only from 120° to 60°, so that one-third of the heat input is entirely wasted. It is therefore usual for such systems to incorporate a re-circulating arrangement by which air from the building is returned to the system for re-heating. This is usually effected by including a damper at the inlet, which can be set to give any proportion of fresh to re-circulated air. When 100 per cent. re-circulation takes place the pressure in the building is lost and infiltration will then occur, and so it is common practice to allow a maximum of 75 per cent. of air to be returned to the system.

Strong winds have the effect of counteracting the minute internal pressure generated by a Plenum system, so that on the windward side of a building air leakages inward through windows, etc., may still occur, with a corresponding drop in temperature.

Owing to the dirt brought in by the fresh air stream an air filter is generally interposed with the object of cleaning the air supply.

The system is particularly suitable, and has been much used, for the heating of factories, owing to its economy of installation, absence of apparatus at floor level, and shortness of time lag.

For the warming of offices or domestic types of building it is, however, not to be recommended. The continuous operation of the fan is necessary for distribution of the heat, and this may form a large part of the running costs. A heated air supply to rooms occupied by sedentary workers often conduces to a state of lethargy or lifelessness which is sometimes so oppressive as to cause headaches. Apart from these considerations there is often difficulty in accommodating large air ducts in a building, and it is for these reasons that the system has found little favour except in the case of factories and similar open space type of building.

Direct-Fired Plenum System—A variation of the Plenum system, used for large space heating such as hangars, is one in which the air is heated by direct firing. The heater takes the form of a series of tubes or passages through which the gases pass. It is economical in first cost and simple to run.

Ventilating Systems—A ventilation system is not truly a heating system, and is mentioned here only to point out differences from the Plenum system.

When a building is ventilated mechanically it has a fan, heater, filter, and duct system similar to that described above, but the air is warmed only to room temperature or a few degrees below and the warming of the rooms is accomplished by direct radiators, panels, or other system entirely independent of ventilation.

It is necessary to remove the air from the room, and a system of exhaust ducts and fans is provided for this purpose.

The great advantage of such a combined system is that a feeling of freshness can be maintained in the rooms, and the objectionable features of a heated air supply are avoided.

HEATING SYSTEMS—GENERAL

The above summary of the various heating systems is believed to cover every type used up to the present time.

Some of the systems are obsolescent and merit no further description, while others, notably the hot-water heating systems and the ventilation systems employing conditioned air, are of increasing importance and will be considered in detail as to design, initial cost, fuel and running cost, etc., in the following chapters.

FUELS AND METHODS OF USE

To complete the discussion of the classification given in Table XVII consideration requires to be given to the various fuels which can be used to supply the energy for indirect systems.

The table shows that the subdivisions of Indirect Systems under the headings (fuel, etc.) on the left-hand side, provide an alternative classification since each of the methods of extracting the heat from the fuel can be used in conjunction with practically any of the methods (already described) for using it.

It is not proposed to discuss this alternative classification at this stage, but to consider each of the various fuels and the boilers for burning them, under separate headings later.

CHAPTER V

Boilers and Combustion

In considering various types of boiler, and the fuels used in them, it is not proposed to spend much time on those types designed primarily for raising steam for driving engines or for process work. More detailed description can be found in various mechanical textbooks.

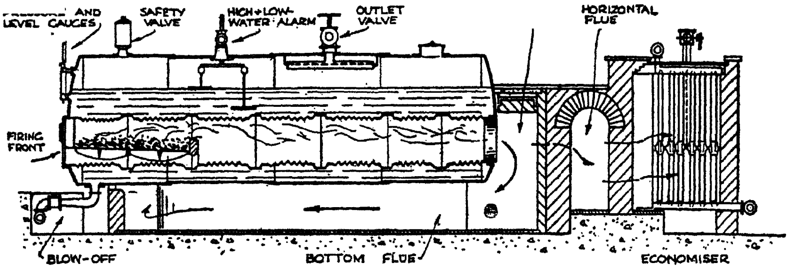
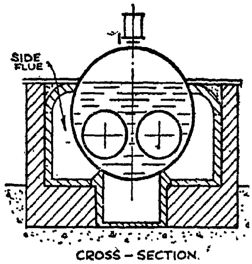


FIG. 45.—Lancashire Boiler for Steam-raising at Medium and High Pressures.



Some of them are, however, referred to briefly as they are in certain cases used for heating, generally as steam generators, but in some cases for heating water.

Lancashire Boilers (Fig. 45) consist essentially of a horizontal steel shell up to 30′ long and 9′ 6″ diameter with flat ends stayed by gusset plates, or with dished ends unstayed, and two horizontal flues below the centre line, generally about 2′ 6″ to 3′ 6″ diameter, in which the grate is provided near the front.

The gases travel to the back, pass under the boiler to the front in brick 'settings', and return to the back in side flues, all with the object of obtaining additional heating area, and of extracting more heat before releasing the gases to the chimney. Even so, the discharge temperature is often 800° or 900° F., which means reduced efficiency and a danger of cracking flues, chimneys, etc.

Hence, an 'economizer' for heating feed water is usually interposed between boiler and chimney, and this may bring the gases down to about 300° to 400° F., whereby the efficiency is much increased. The economizer is either of vertical tube type or high velocity gilled or 'H' type. The former consists of a stack of pipes about 4 in. diameter provided with scrapers worked mechanically for keeping the outside clean of ash deposit

and soot. The high velocity type is more compact; it is cleaned by soot blowers.

The boiler plates are generally ¾ in. to 1 in. thick on outer shells and ½ in. to ¾ in. on flues, but may be thicker for high pressures. The flues are sometimes corrugated to reduce the stresses otherwise caused by their greater expansion acting on the ends of the boiler.

The advantages are:

(*a*) Low first cost.
(*b*) Simplicity.
(*c*) Ease of cleaning.
(*d*) Long life.
(*e*) Big steam capacity, useful with a fluctuating load.

The disadvantages are:

(*a*) Great space required.
(*b*) Long time lag.

Maximum pressures are normally up to 250 lbs. per sq. in. Heating capacity about 2½ million to 8 million B.T.U.'s per hour with natural draught, and up to about 16 million with forced and induced draught and economizers.

A variation of the Lancashire type is the *Galloway Boiler*, which is provided with a number of diagonal cross-tubes in the flues.

Cornish Boilers are similar to the Lancashire type, but are generally not more than 24 ft. long and 6 ft. in diameter. Their distinguishing feature is the use of a single instead of a double flue. They are historically interesting as having been first employed to drive engines in the Cornish tin-mining industry over a hundred years ago, when the first beam engines were introduced. In those days their working pressure was about 5 to 10 lbs. per sq. in., but to-day they can be made for pressures up to 200 lbs. per sq. in. Their heating capacity may be up to half that given for Lancashire boilers.

An improvement on the Cornish boiler is the Cornish Trentham, which has additional heating surface provided in the flue by means of cross tubes.

'Economic' Boilers—These are similar to the types just described in having a horizontal flue or flues near the bottom of a horizontal shell. They differ, however, in being shorter (generally not more than 14 ft.) and in bringing the gases back to the front in a series of fire tubes.

The gases can then be alternatively:

(*a*) Collected at the front and go to stack.
(*b*) Returned to the back in a further stack of fire tubes (Fig. 46).
(*c*) Returned to the back outside the shell in brick settings (Fig. 47).

(*a*) and (*b*) have the advantages of not needing settings, (*b*) giving the highest efficiency. This type with the (*b*) alternative is used for water heating in the Bank of England, and gave under test the remarkable

efficiency of 89 per cent., but the conditions here are unusually favourable in having:

(*a*) Low water temperature.
(*b*) Exceptionally efficient lagging.
(*c*) Absence of forcing.
(*d*) Very low discharge gas temperatures.
(*e*) Oil firing.

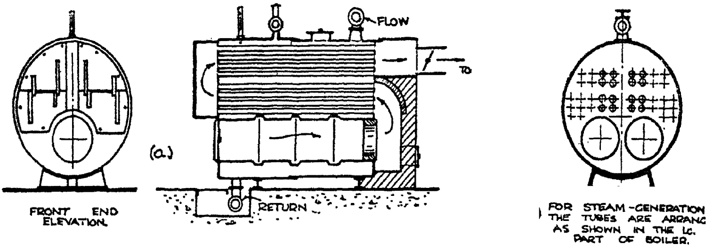


FIG. 46.—Return Tube Boiler; (*a*) for Water; (*b*) for Steam.

'Economic' boilers are used both for steam and high or low pressure hot water, and have the advantages, as compared with Lancashire or Cornish boilers, of:

(*a*) Saving of space and weight for a given capacity.
(*b*) Lower time lag.
(*c*) Greater efficiency without economizers.

They require frequent or continuous stoking, whether hand or mechanical, and regular cleaning of tubes. They can be constructed for pres-

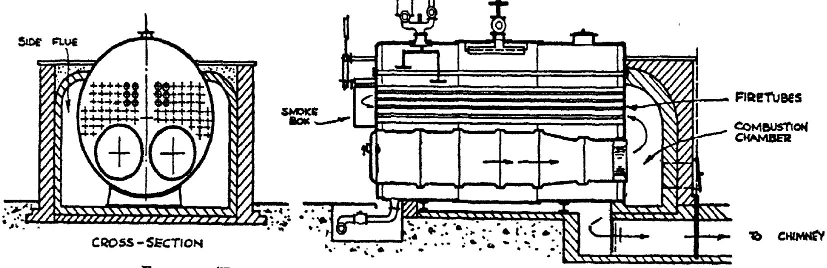


FIG. 47.—'Economic' Type Boiler for Steam-raising at High Pressures. For Water Heating the Steam Space is filled with Fire Tubes.

sures as for Lancashire type, and have a wide range of heating capacity from 800,000 to 11,000,000 B.T.U.'s per hour with natural draught and 50 per cent. more with forced or induced draught and automatic stokers.

A recent example of 'Economic' boilers is shown in Plate IV.

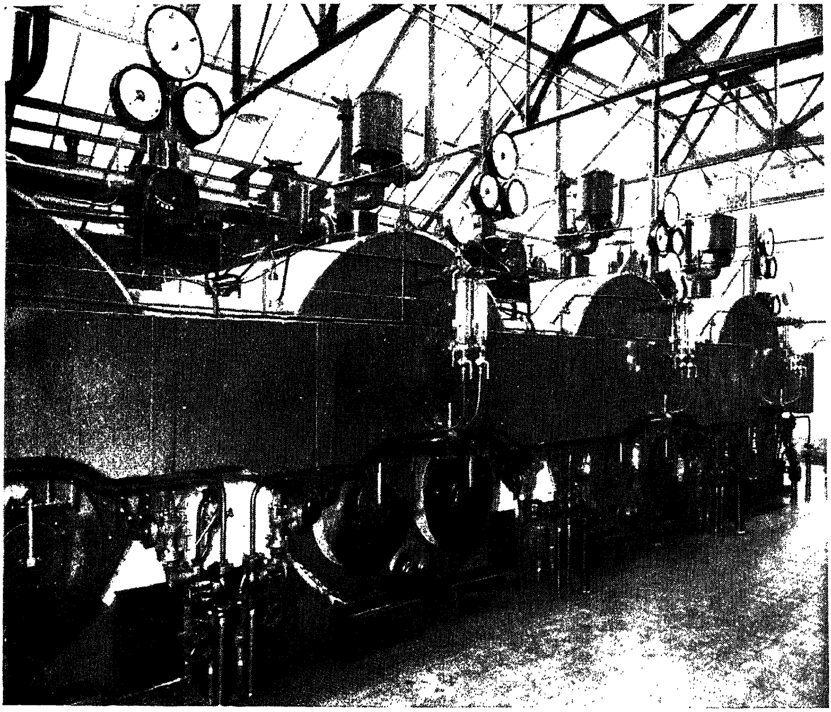


Plate IV. A typical installation of 'Economic' boilers, arranged for oil firing (see p. 98)

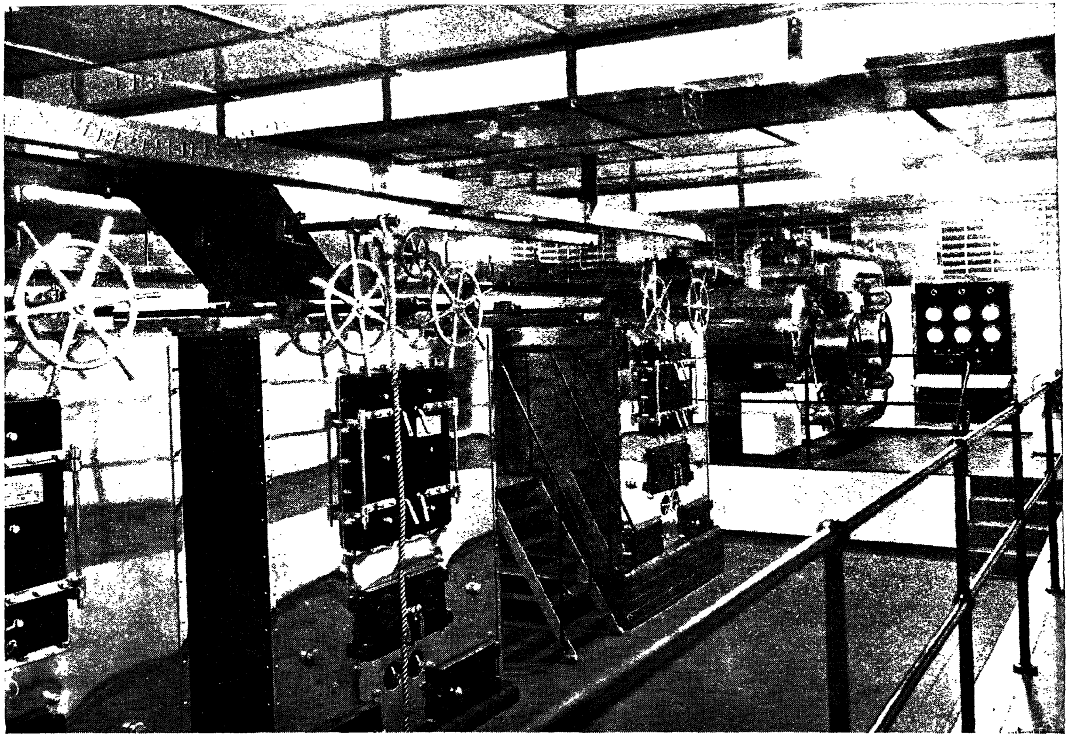


Plate V. Example of boiler house with magazine coke boilers—Gas, Light & Coke Co., London (*see p.* 101)

Super Lancashire Boilers are a recent development and improvement of the Lancashire, having two large flues near the top of the shell and a stack

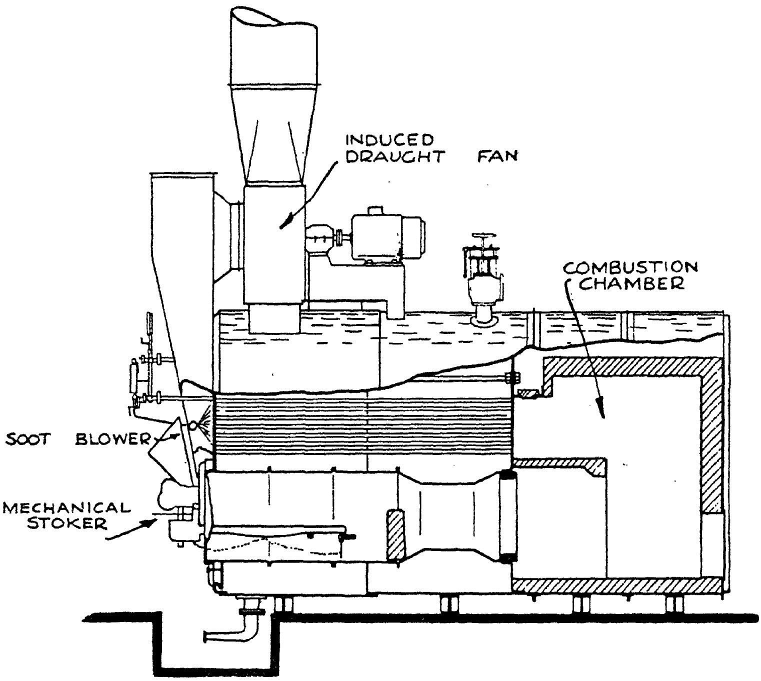


FIG. 48.—Cochran-Kirke 'Super-Economic' Boiler with Induced Draught.

of fire tubes near the bottom, and, in addition, a further battery of tubes at each side outside the shell for pre-heating the air for combustion.

They have advantages over the 'Economic', and a higher efficiency, due to the preheating, especially useful when a cheap fuel is being burned with forced and induced draught, which are both necessary with this type.

'Super-Economic' Boilers are a still more recent development, of which there are several versions. One type is illustrated in Fig. 48 and is notable for the special form of fire tube which is used. This is of a wave shape, calculated to give a scouring action to the gases with resulting higher efficiency. The boiler is also provided with a very large firebrick-lined combustion

SECTION OF "SINUFLO" TUBE

chamber at the back, an induced draught fan and a mechanical stoker. The combination of these features is stated to give smokeless combustion with most fuels and efficiencies of 80 per cent. or over. Due to the fan, a tall chimney is unnecessary.

Another type of Super-Economic boiler is shown in Fig. 49. This has a double return tube arrangement with the gases leaving the boiler at the bottom, which is the coldest part. Thus, high heat transfer is obtained and the gases leave at about 350° F. giving high efficiencies. The boiler works with an induced draught fan, and is usually fitted with a mechanical stoker.

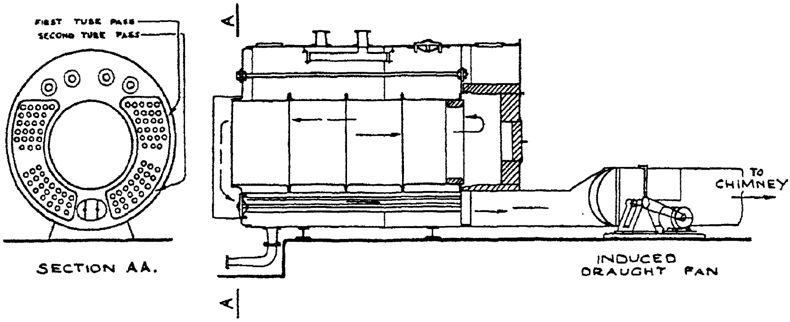


FIG. 49.—'Super-Economic' Boiler with Induced Draught. (Danks of Netherton, and John Thompson.)

Loco. Boilers have a large fire box inside the main shell, and, from the back of the box, the gases pass through a number of fire tubes to the smoke box at the farther end, from which they pass straight to stack.

These boilers give a high capacity in a small space, because the gases emerge at a high temperature, but they are correspondingly inefficient and seldom used for heating in permanent installations.

Water Tube Boilers are exemplified by many well-known types, to which we need not refer by name. They are chiefly used for power station work, but are also suitable for large-scale heating systems for steam or high pressure hot water in units of from 10 to 50 million B.T.U.'s per boiler. Their characteristics are:

Water is contained within the tubes, the hot gases being outside.
Short time lag, hence rapid response to change of load.
High efficiency.
Adaptability for the burning of any type of fuel.
Compact size for large outputs.

The Water Tube Boiler requires more skilled attention and greater care in the treatment of feed water than the Lancashire type, thus in institutions, hospitals and many industrial plants the latter is still preferred on account of its reliability.

Several specialized types of Water Tube Boiler of forced circulation

type have been developed. Their main interest here is in their application to High Pressure Hot Water systems, and they are dealt with under Chapter XV on this subject.

Vertical Boilers vary from the simple types to the more developed ones. The former consist of a fire box contained in a vertical shell with one to six large water tubes crossing the fire box, which terminates at the top with a vertical funnel passing through the steam space at the top of the vertical shell. They are much used by contractors for steam cranes, pile-drivers, etc., also for dairies, kitchens and other relatively small steam supplies. They are cheap, but have a low efficiency and are seldom used for water heating. Heating capacities up to 2,000,000 B.T.U.'s per hour can be obtained.

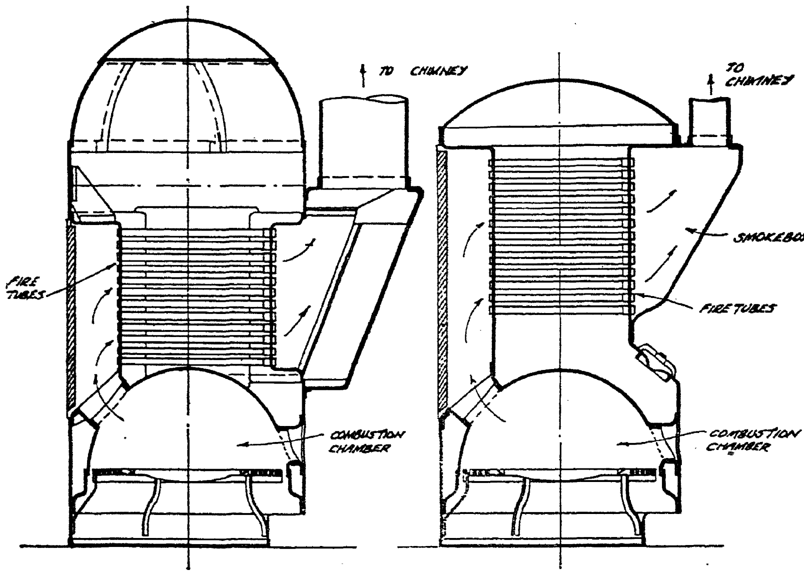


FIG. 50.—Cochran Boilers, Vertical Type; (*a*) for Steam; (*b*) for Hot Water.

The more highly developed type, as in Fig. 50 (*a*), has a large number of horizontal *fire* tubes and no water tubes. The hot gases leaving the fire box go to the outside of the shell and return through the fire tubes before passing to the stack. Much higher efficiencies are obtainable, and this type is suitable for heating capacities up to 6,000,000 B.T.U.'s per hour, at which rating the size is 19 ft. high by 9 ft. diameter.

The same design without the steam dome is produced for water heating as in Fig. 50 (*b*).

Vertical boilers of this or other types need great height but small floor space, and this is occasionally an advantage, as in a congested works yard, kitchen, or laundry.

BOILERS SPECIALLY DESIGNED FOR HEATING

Cast-Iron Sectional Boilers—The term 'cast-iron sectional boiler' is now almost a household word; and many well-known types will at once occur to any reader. They consist essentially of a series of sections, the waterways of which are connected by cast-iron nipples, generally in three places.

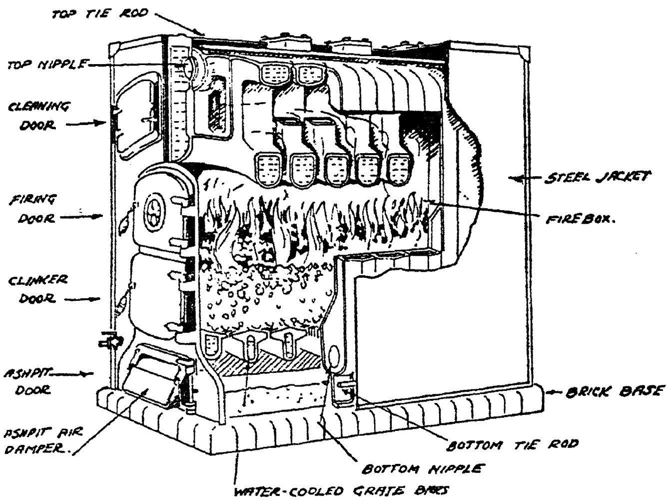


FIG. 51 (*a*).—Cast-iron Sectional Boiler ('Ideal').

Separate bolts running from front to back (Fig. 51 (*a*)), or in another make separately between the sections (Fig. 51 (*b*)), keep the joints water-tight between the sections and nipples.

Each section normally consists of two legs, forming the sides of the fire box, and a top portion bounding the sides of the flues to return the gases to

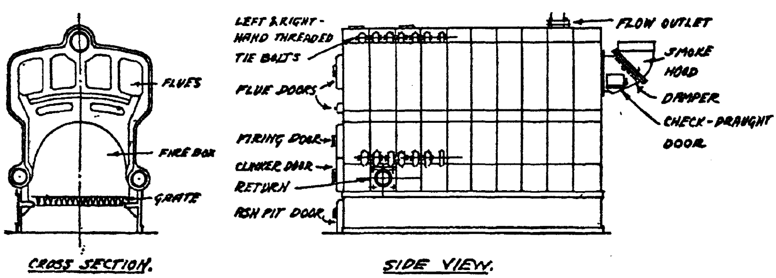


FIG. 51 (*b*).—Cast-iron Sectional Boiler ('Robin Hood').

the front (secondary surface) and again to the back (tertiary surface). The front and back sections differ from the intermediates in having waterways over the greater part of the area, except where openings for doors or smoke outlet occur.

In some types the sections are in halves, with the return flues at the sides, and in this case each half has two nipples, making four in all.

Fire bars generally consist of loose castings laid near the bottom, but in one particular make they are water-cooled, so obtaining additional heating surface, reduced clinkering and absence of burning of the bars.

Doors on the front are provided for feeding the fuel, clinkering, ash removal and for cleaning the flues.

The ash door forms the air inlet under the grate, and for this reason has either a hit-and-miss regulator to be set by hand, or a hinged damper controlled by a simple direct-acting thermostatic regulator in the top of the boiler. The latter is generally connected to the ashpit damper with a chain (as in Fig. 52), having an adjustment for various temperatures.

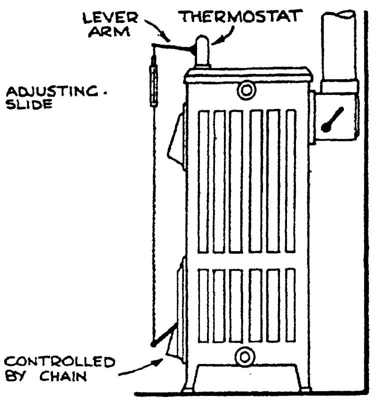


FIG. 52.—Damper Regulator, showing Method of Fitting to Small Heating Boiler.

This simple control will in many cases effect considerable savings in fuel, but it is not a perfect device as its effect is upset by such factors as wind, fuel bed thickness, amount of clinker, and flue temperature.

The fuel door usually has an adjustable regulator fitted for the admission of secondary air above the fuel bed to burn the carbon monoxide which would otherwise escape unburnt to the flue. This is particularly necessary with anthracite, as, without it, combustible gases may completely fill the boiler and chimney and become suddenly ignited with a small explosion when the flame eventually burns through.

Secondary air is more effective if pre-heated before admission, as, when cold, it may only serve to cool down the gases in the fire box and retard combustion. For this reason one maker introduces this air through a passage between an inner and outer plate of the clinker door, which is adjacent to the fire and therefore always hot.

The back of this type of boiler is fitted with a smoke outlet hood, consisting of a large casting with openings to which the flue pipe is connected, either vertically or horizontally. It also contains a damper operated by a rod from the front, and a check-draught door. The latter, when opened, allows cold air to enter the flue and serves the double purpose of cooling the outlet gases and reducing the draught in the boiler.

It is necessary for economy of fuel to insulate this or in fact any type of boiler, and the more efficient the insulation the less heat is wasted in the boiler house. This insulation takes a variety of forms, and is referred to on page 148.

It is usual to mount cast-iron sectional and other similar boilers on a brick base, usually blue brick with bull-nosed edge. The purpose of this base is to raise the boiler above floor level so as to prevent water on the boilerhouse floor from collecting around the metal of the boiler where it would cause rusting and deterioration. The raised base also allows the ash-pit door to be swung open freely without catching on lumps of clinker, fuel, etc., on the floor.

The base is usually $4\frac{1}{2}$ in. high, 9 in. wide, and the centre is preferably filled with firebrick, sunk towards the back. This gives a smooth surface for withdrawal of ashes, and is not liable to cracking. With oil firing it is desirable to insert a layer of insulating brick under the firebrick lining.

The general characteristics of the cast-iron sectional boiler may be summarized as follows:

(1) Low in cost for moderate sizes.

(2) Life of ten to twenty years, depending chiefly on degree of forcing, pressure, temperature variation, and whether kept clean when not in use. When forced, the metal is liable to burn in time and lose its strength.

(3) The boilers are not to be recommended for water-pressures exceeding 100 ft. head or 120 ft. as an outside maximum (including pump head) because the metal gradually yields plastically, with eventual failure by cracking. Thus whilst, when new, these boilers may be satisfactorily tested to 100 lbs. per sq. inch (230 ft. head) or more, in the later years of their life they may be unable to withstand half this pressure.

(4) Rapid variation in temperature of inlet water is liable to crack the sections by sudden contraction stresses, hence these boilers are not suitable for direct hot-water supply (except when specially designed for this purpose).

(5) When not cleaned and left well ventilated through the summer season, the residual ashes and soot attract moisture, which forms sulphurous and sulphuric acid, which in turn corrodes the metal. Many boilers shorten their life much more from this cause in summer than from all the causes which operate when in use. This effect is, of course, common to all types of boiler, but its results are probably more serious with cast iron, since with this material the thickness of metal cannot be reduced without failure as it can with steel or wrought iron.

(6) More compact for a given rating than most boilers.

(7) More adaptable to meet an increase in duty by adding sections and to fit into any available space. Small boilers may have as few as four sections, and this may be increased to fourteen or more when extensions are required. As sections are added each carries its own portion of grate area, so maintaining a correct balance with the heating surface.

(8) Not so efficient as some types owing to the large cross-sectional area of the flues allowing gases to escape at high temperature when working at full rating. This is a consequence of the makers being obliged to design the boiler to work under the poorest conditions of draught, such as may apply,

for instance, for greenhouse heating, where perhaps only a 15 ft. chimney may be available.

(9) Suitable for steam at low pressure (up to 15 lbs. per sq. inch), provided the whole of the condensate is returned so that no incrustation occurs; it is not possible to clean these boilers internally.

(10) The small gaps which exist between the sections should be closed with boiler stopping to prevent air leaks, and similarly the doors should fit well, otherwise air leaks will occur, reducing efficiency and preventing proper control. Machined surfaces are essential for good results.

(11) It is usual to make the volume of the fire box much larger than is in fact essential. This is to allow a fuel capacity sufficient for a run of several hours during the day time at full load, and eight to twelve hours at night on a third or quarter load without stoking. Whilst this is a desirable feature from the point of view of the frequency of attention otherwise required, it is the cause of much wasteful firing. When a large mass of black fuel is suddenly placed on a bright fire volatile gases and carbon monoxide are generated and escape up the chimney unburnt. This may go on for several hours until the top of the bed becomes incandescent, by which time as much as one-third of the heating value of the new fuel may have been lost. In the meantime the water temperature will probably have dropped seriously, and the boiler then has to be forced to make this up.

It is this factor which accounts for wide variations which occur in the fuel consumption between one building and another. The skilful man will always try to keep a bright fire by giving regular attention during the day, whereas the careless man will load on the fuel whether it is necessary or not. At the same time the skilled stoker will be giving greater satisfaction to the building occupants by reason of the steady temperature maintained.

For best results, the fuel bed should be about 8 in. thick, uniform and without holes, but this involves frequent stoking, and about 12 in. thickness is more generally adopted. On the other hand, if the stoking is too frequent, loss of efficiency results from the admission of large quantities of cold air through the open firing-door. Hand-stoking is entirely a matter of skill, and its efficiency is largely bound up in the question of fuel bed thickness and draught.

Boilers of the cast-iron sectional type have ratings from 20,000 to 2,000,000 B.T.U.'s per hour, a very wide range. Where greater outputs are required, two or more boilers are connected together in a 'battery'. Indeed, for any rating over about 800,000 B.T.U.'s per hour, it is considered good practice to provide two boilers each capable of at least two-thirds of the net full-load duty. This arrangement gives a measure of standby-capacity and makes for easier control in mild weather.

The cast-iron sectional type of boiler only became possible as a result of wonderful perfection in casting and coring hollow sections without blowholes, spongy patches or dangerous cooling stresses.

Wrought-Iron or Steel Sectional Boilers—Early heating boilers were always of wrought iron, made in one piece, generally of semi-circular section, and were known as 'saddle' boilers. These were built into a brickwork

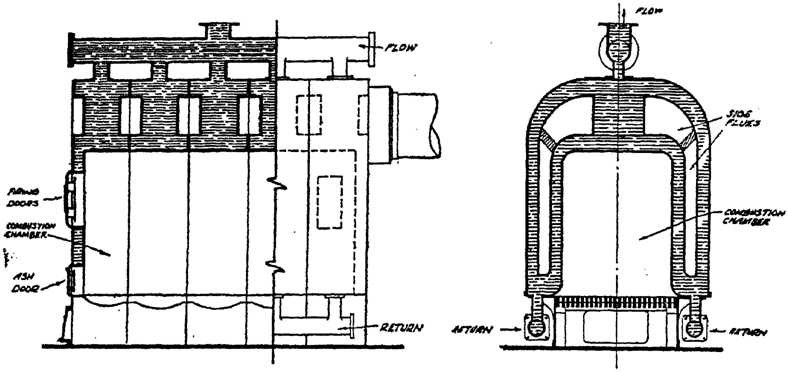


FIG. 53.—Typical Steel Sectional Boiler (Paxman).

setting and, owing to their small heating surface and the radiation from the brickwork, were low in efficiency.

With the introduction of the cast-iron sectional boiler, early in the

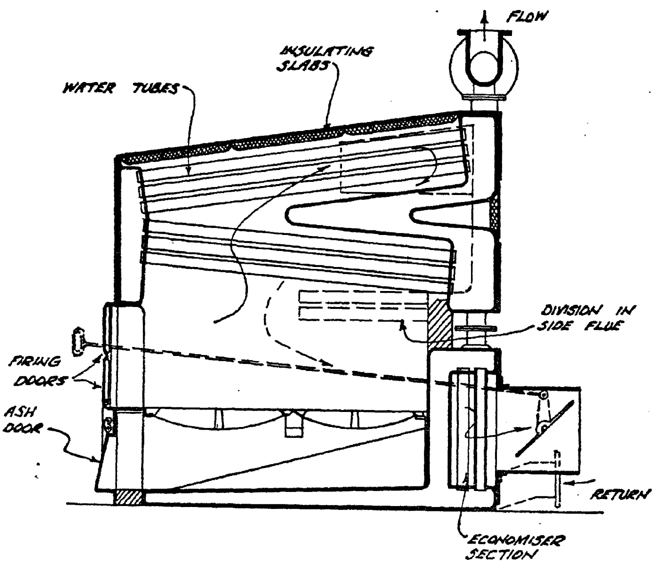


FIG. 54.—Steel Sectional Boiler, Water-Tube Type ('Metropolitan').

present century, the old saddle type fell into disfavour and has now become obsolete.

The disadvantages of cast iron, namely, inability to withstand the high pressure head of lofty buildings and comparative shortness of life have caused wrought iron to come into use in a new form, the steel sectional boiler. This possesses the advantages of the cast-iron type as regards adaptability, efficiency, and ease of handling in confined spaces, with the permanence and strength associated with wrought metal.

The seams are joined by electric welding, and nowadays mild steel is almost invariably used, as this is more easily welded than wrought iron, apart from which the latter is not produced in sufficient quantity nor at a price low enough to meet present needs.

A great variety of types is now produced, of which two are illustrated in Figs. 53 and 54.

Fig. 53 shows a type which is a copy in steel of the cast-iron pattern and differs chiefly in the width of the sections, which are generally about 15 in. as compared with 7 in. of the cast-iron, and in having surfaces without convolutions owing to difficulties of manufacture. The size required for a given output is therefore greater.

Fig. 54 shows another type employing water tubes, and differing in form entirely from the cast-iron design.

The tubular heating surface is of great efficiency and the greater portion of it is almost self-cleaning. The sections are in rather larger units than other types and do not allow of the same facility for extension.

Steel sectional boilers of either type have all the characteristics already mentioned for the cast-iron sectional, but differ principally as follows:

(1) More costly.

(2) Longer life (thirty to fifty years).

(3) Less liable to be affected by burning of plate, change of water temperature, forcing, etc., because the metal is ductile instead of brittle and is also thinner.

(4) Suitable for working pressures up to 100 lbs. per sq. in. or 230 ft. head of water, and for steam pressures up to 25 lbs. per sq. in.

(5) Capable of higher efficiencies as the technique of manufacture permits more flexibility in design, so that smaller flue passages can be arranged for if draught conditions permit.

(6) External headers are used for connecting the waterways of adjacent sections instead of nipples.

(7) Sizes are available from 100,000 to 5,000,000 B.T.U.'s per hour.

Magazine Boilers—The disadvantage of all the boilers already described as specially designed for heating is that they require frequent stoking, and it was not until this defect had been remedied by Continental and American designers that magazine boilers were introduced into this country. (It should perhaps be stated that one of the authors first saw boilers of this type

in Athens and in Denmark many years ago.) They are now deservedly finding favour in this country.

Magazine boilers have the following advantages over others:

(1) Need filling once in twelve or twenty-four hours only, and are self-regulating under thermostatic control between fillings.

(2) Have a large grate area, thus keeping down the temperature of the hottest part of the fire so as to minimize clinkering.

(3) Provide constant thickness of fuel bed, and therefore high average efficiencies.

(4) Dirt and dust can be avoided by the use of proper fuel-feeding arrangements.

(5) Absence of moving parts.

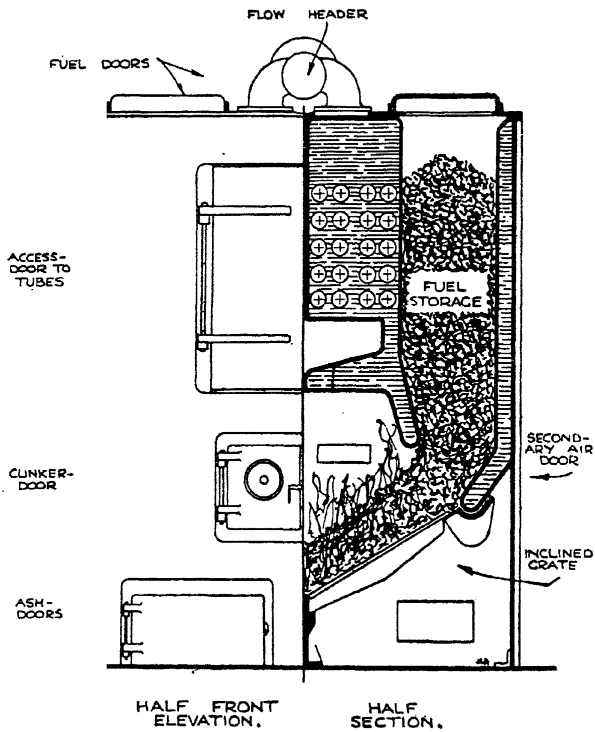


FIG. 55.—Steel Magazine-Type Boiler ('Gravico').

By reason of these factors, magazine boilers reduce labour and attention to small proportions, so that even though the first cost is higher, the running cost is much less. They give most of the advantages which can be

secured with oil, and, in addition, use fuel of a lower cost and one that is British in origin.

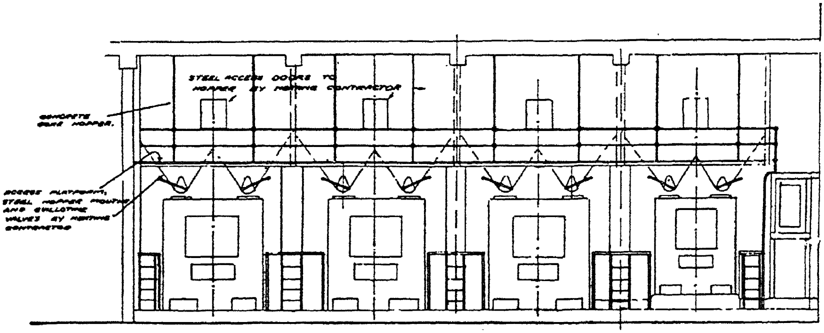


FIG. 56 (*a*).—Details of Boiler House, showing Magazine Boilers fed from Hoppers Overhead. Front View.

One successful type of magazine boiler made of welded steel is shown in ·55·

Magazine boilers can and should be so arranged that fuel is filled

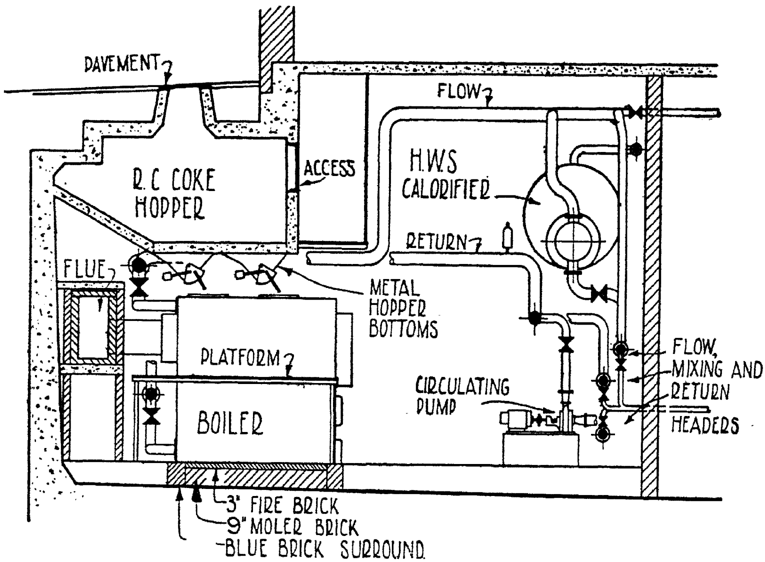


FIG. 56 (*b*).—Cross-section of Fig. 56 (*a*).

straight into the top without any intermediate labour. A simple method of accomplishing this is to place the boilers under the pavement with a coal-plate feeding into a hopper over each, as in Figs. 56 (*a*) and (*b*).*

Where this is impossible, a simple system of hopper and conveyor is

*See also Plate VIIIA, p. 157.

desirable to effect the same purpose with a minimum of handling. This arrangement is shown in Fig. 57 and in Plate V (facing p. 99).

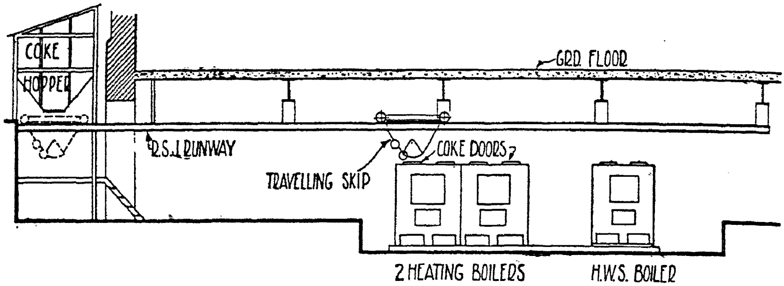


FIG. 57.—Typical Longitudinal Section of a Boiler House, showing Magazine-Type Boiler fed by Travelling Skip from Hopper at side.

Ash removal is only needed once daily, and generally occurs at the time of fuel filling.

At the moment, the rated capacities vary from 100,000 B.T.U.'s per hour to 2,000,000.

COMBUSTION OF FUEL*

Before the various methods of firing can be discussed it is necessary that the principles of combustion should be understood.

Fuels encountered in practice are mixtures or compounds chiefly of carbon, hydrogen and oxygen. In addition there are generally small quantities of sulphur, nitrogen, and in the case of solid fuels—ash.

A proportion of the carbon, hydrogen and oxygen are usually combined in an organic form classed under the general label of '*Volatiles*' (chiefly hydrocarbons). These are driven off in the early stages of combustion and give rise to white, yellow or orange flame. If cooled too quickly on the water-backed surfaces of the boiler, or by too free an admixture of cold air above the fire bed, they often give rise to smoke. The higher the proportion of hydrocarbons in the fuel, the longer and more lasting the flame. After the hydrocarbons have been consumed, in the case of coal, the mass of fuel remaining is largely coke. The carbon in the hydrocarbons burns to carbon dioxide (CO_2) and the hydrogen (H_2) with oxygen (O_2) becomes water in the form of superheated steam.

The uncombined carbon or *Fixed Carbon* is that portion of the fuel which exists as pure carbon. In the case of coke, the greater part is fixed carbon. It is considered that the first stage of its combustion is to carbon monoxide due to the restricted air supply in the fuel bed, and this is then burnt into carbon dioxide at the top of the bed, giving rise to a short blue flame.

*The subject of combustion is highly involved and in some cases controversial. General principles as affecting heating boilers only can here be touched on.

The hydrogen not combined with carbon is combined with oxygen, as water, referred to as *Inherent Moisture*. Solid fuels generally also contain *Free Moisture*, due to their hygroscopic nature, and due to rain in open trucks, etc. The latent heat of evaporation has to be supplied from the heat produced, and this is not recovered in the practical boiler, in which the gases leave above 212° F. If the gases leave below this temperature, part condensation will occur, and part of the latent heat will be recovered.

The *Sulphur* on combustion forms sulphur dioxide (SO_2), trioxide (SO_3), or hydrogen sulphide (H_2S).

The combustion of fuel is a physico-chemical process which is accompanied by the liberation of heat. Combustion reactions can only take place at a high temperature known as the ignition temperature, which varies between 700° and 1000° F. according to the fuel.

The elements combine with oxygen in proportion to their molecular weights, which are as follows: $O_2=32$, $C=12$, $H_2=2$, $S_2=64$, $N_2=28$. Air contains 23·15 per cent. of oxygen by weight and 76·85 per cent. of nitrogen, etc. Thus

TABLE XVIIIA

Reaction	Products	Weight of O_2 per lb. Combustible	Equivalent Weight Air lbs./lb.	Heat Liberated B.T.U./lb.
$C+O_2$ $12+32$	$=CO_2$ $=44$	$2\frac{2}{3}$	11·6	14,600
$2C+O_2$ $24+32$	$=2CO$ $=56$	$1\frac{1}{3}$	5·8	4,350
$2CO+O_2$ $56+32$	$=2CO_2$ $=88$	·57	2·5	4,370
$2H_2+O_2$ $4+32$	$=2H_2O$ $=36$	8	34·5	62,000
S_2+2O_2 $64+64$	$=2SO_2$ $=128$	1	4·34	3,930
*Methane** CH_4+2O_2 $16+64$	$=CO_2+2H_2O$ $=80$	4	17·4	23,900
*Ethylene** $C_2H_4+3O_2$ $28+96$	$=2CO_2+2H_2O$ $=124$	3·4	14·9	21,050

* These occur in towns' gas.

The Calorific Value of a fuel is the quantity of heat released on the complete combustion of unit weight. There are two such values always given—gross or higher, and net or lower. The *gross calorific value* includes the heat given up in the reaction to supply the latent heat of vaporization to the water, which forms part of the products of combustion, as explained above. The *net calorific value* is the gross value minus the latent heat referred

to. The greater the amount of hydrogen in the fuel the greater the difference between the two values. In the case of oil the difference may be about 1200 B.T.U.'s/lb. out of 19,600 (gross), equal to 6 per cent.; in the case of gas 60 out of 500 (gross), equal to 12 per cent. With bituminous coals the difference is less, with anthracites still less, and with coke almost nil.

The calorific value of a fuel may be calculated approximately from a knowledge of its analysis and the heat due to the reaction with oxygen of each element. With a fuel, say, 75 per cent. carbon, 5 per cent. hydrogen, 20 per cent. ash moisture, etc., the heat value will be (using Table XVIIIA)

$$(\cdot75 \times 14{,}600) + (\cdot05 \times 6{,}200) = 11{,}260 \text{ B.T.U.'s/lb. (gross).}$$

Such a result cannot be accurate as it depends, to some extent, on the chemical combinations existing in the fuel, and it is necessary to check any such computation by experimental determination. For this purpose a *bomb calorimeter* is used, the Mahler Donkin type being the standard instrument. (See B.S.S. Nos. 845/1939, 735/1937, 687/1936 and 435/1932.) The bomb is simply a closed steel vessel. In this a weighed quantity of dried fuel is placed in a crucible, and is then filled with oxygen under pressure. The bomb is contained in a calorimeter filled with water with stirrers and a finely graduated standard thermometer. The fuel is ignited by electrically-heated platinum wire dipping into the crucible. When the combustion is completed the rise in temperature is observed, and from this and a knowledge of the heat loss from the calorimeter, the calorific value may be determined. In the British units this is expressed in B.T.U.'s per lb.

For gas, a Boys calorimeter is generally used. In this, gas is burnt at a known standard rate in air, and the amount of heat given out is determined by the rise in temperature of water passing through the calorimeter.

Fuel Analysis—The terms *proximate* and *ultimate* require explanation.

The *proximate analysis* gives the percentage by weight of the fixed carbon, volatile matter, moisture and ash. From this the probable characteristics and form of the combustion can be forecast.

The *ultimate analysis* gives the percentage by weight of the various elements or compounds contained in the sample. From this the theoretical air required for combustion may be estimated. The combustion efficiency may also be arrived at, given a knowledge of the flue gas analysis.

The following table gives some typical proximate analyses and calorific values for various solid fuels : oil and gas are dealt with under their separate chapters later.

Combustion Losses—Heat losses occur in combustion owing to the gases leaving the boiler at a temperature higher than that of the fuel and air prior to combustion.

This represents hot gases going to waste, the amount of such waste being proportional to:

the wt. of gases × rise in temperature × specific heat of the gases.

TABLE XVIIIB

PROXIMATE ANALYSIS OF SOME TYPICAL SOLID FUELS

Coal	Proximate Analysis				Calorific value as fired		Characteristics
	Fixed Carbon, %	Volatile Matter, %	Total Moisture, %	Ash, %	Gross	Net	
Poor Slack, Leicester	41·5	30·0	10·6	17·9	9,660	9,220	Non-caking, Shaly
Singles, Warwickshire	49·3	34·0	9·8	6·7	12,020	11,520	Long Flame, Caking
Singles, North-East	55·4	34·0	6·8	3·8	13,560	13,030	Bituminous, Strongly Caking
Blended Smalls, South Wales	72·4	16·7	3·2	7·6	14,020	13,950	Semi-Bituminous
Welsh Steam Coal	82·0	12·0	1·9	4·1	14,500	14,300	Anthracitic
Anthracite	91·0	5·0	2	2	14,800	14,650	—
Coke (Gasworks), Broken	84·3	2·9	7·0	5·8	12,700	12,450	—
Coke (Gasworks) Breeze	70·0	0·5	12·9	16·6	11,000	10,000	—
Coke (Coke Oven)	91·0	1·0	4·0	4·0	13,600	13,400	—

If more oxygen is supplied than is essential for complete combustion, the weight of gases is increased and hence the losses.

In any case, as air only contains about 20 per cent. of oxygen, a considerable amount of flue gas must go away as nitrogen, which involves an unavoidable loss. The heat loss can be reduced to a minimum by:

(*a*) Reducing the air admitted to the lowest amount which will provide complete combustion.

(*b*) So arranging the boiler that the gases will escape at as low a temperature as possible.

There are, however, practical limits to both (*a*) and (*b*).

As regards (*a*), unless there is some excess of air, combustion will in practice be incomplete, and the flue gases will contain solid fuel (in the form of smoke), and some unburnt gases in the form of hydrocarbons and CO. A compromise therefore has to be effected, and in practice it often happens that a chimney with slightly visible smoke may involve lower losses than one which is clean, as this generally means a considerable excess of air. This applies to coal. Coke does not emit smoke.

As regards (*b*), it is not practicable to discharge below 212° F. because

of condensation troubles (and resultant corrosion in the boiler), and because in practice it would involve very large boiler surfaces to get down to such temperatures. As a rule, the outlet temperature is nearer 400° to 600° F. Some of the heat may, however, be recovered by using it to preheat the air used for combustion, but this is only done in very large plants.

Flue Gas Analysis—It is clear that as the air is reduced towards the minimum required for combustion, the CO_2 content of the flue gases will rise, and this is used as a guide in boiler-house operation. Hence a CO_2 recorder is generally fitted for the help of the attendant in large plants.

In round figures the maximum theoretical percentage of CO_2 (by volume) if all the fuel is carbon is 20 per cent., since roughly one-fifth of the air is oxygen.

As, however, part of the fuel is often hydrogen which oxidizes to produce water, and sulphur, which produces SO_2, etc., the maximum theoretical value must be under 20 per cent. by an amount which increases as the carbon content of the fuel becomes less than 100 per cent. of the combustibles. In practice CO_2 percentages between 10 and 15 are satisfactory, depending on the fuel.

The complete analysis of flue gases requires a knowledge of the CO and O_2, the former revealing partly-burnt gases and the latter excess oxygen.

The standard apparatus for analysis is the Orsat apparatus, which consists of three glass vessels containing absorbent solutions with glass tubes or beads for increasing the contact surface. A volume of gas is drawn into a measuring burette with a hand pump, and then exposed to the solutions in turn. The steam in the gases is condensed by a water jacket outside the first tube or by other means.

The first is a caustic soda solution which absorbs CO_2. The unabsorbed portion is then returned to the burette and measured again, when the loss in volume gives the CO_2 content.

The second vessel contains an alkaline solution of pyro-gallol for absorbing, and thus determining the residual oxygen.

The third contains an acid solution of cuprous chloride for determining the carbon monoxide.

CO_2 indicators and recorders for daily use in boiler-houses are, however, now made automatic by several instrument-makers, some depending on caustic soda absorption, some on thermal conductivity (see Fig. 58), some on weight, and some on osmotic pressure.

The CO_2 content alone correctly indicates the amount of excess oxygen in the flue gases, only if a proper correction is made to the theoretical value for pure carbon. Hydrogen produces water on burning, and pure hydrogen would give a CO_2 content of zero, whether there were no excess or infinite excess of air. This is particularly the case with oil, which often contains 15 per cent. of hydrogen, and towns' gas, about 20 per cent. (by weight). The true excess air must, therefore, be measured directly from the flue-gas content. It must be remembered that the CO_2 analysis gives a

proportion by volume not weight, and that the analysis is of the dry gas, the steam having been condensed.

Flue Gas Temperature—A large excess of air does not necessarily mean a low boiler efficiency, because if the gases are discharged at a sufficiently low temperature, most of the heat in the excess air will be given up to the

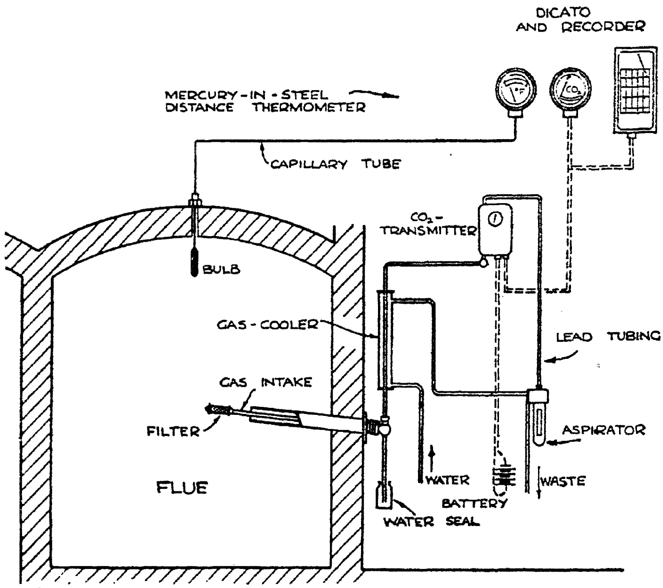


FIG. 58.—Diagram of Electrical CO_2 Recorder and High-Temperature Thermometer fitted to Boiler Flue.

boiler heating surface. Hence the temperature of the gases must be known at the point where they leave the boiler.

This temperature is measured by a pyrometer or high temperature thermometer, which may be of the mercury in steel type, nitrogen-filled mercury in glass, or electrical type.

Theoretical Efficiency—It is interesting to consider the maximum theoretical combustion efficiency with various flue gas discharge temperatures and excess air contents. This can be shown as follows, taking a typical coal analysis.

Coal	Air for Combustion	Lbs. Air for 1 lb. Coal
C.V. 13,000 B.T.U.'s/lb.		
80% Carbon	·8 × 11·6 lbs./lb.	9·28
4% Hydrogen, 1·1% Oxygen *	$\left(\cdot 04 - \frac{\cdot 011}{8}\right)$:	
1% Sulphur (ignored)		
13·3 × Ash, Moisture, etc.		
100%		10·61 lbs.

* The oxygen burns $\frac{1}{8}$ of its equivalent in hydrogen, i.e. $4 - \frac{1 \cdot 1}{8} = 3 \cdot 875$ if hydrogen is left requiring air for combustion.

Products of combustion weigh 10·61 + 1 lb. fuel = 11·61.

Specific heat gas (mean), ·24. 11·61 × ·24 = 2·79 B.T.U./lb. fuel per deg. F.

Loss in products of combustion $= \frac{2·79 \times 100}{13,000} = ·0214\%$ per degree.

Thus for 100° rise, flue loss = 2·14%.

„ 500° „ „ = 10·7%.

The above is with no excess air.

If the air supply is 50% in excess, air weight = 15·915,

$$\text{for } 500° \text{ rise flue loss} = \frac{(15·915 + 1) \times ·24 \times 500 \times 100}{13,000} = 15·62\%.$$

The maximum CO_2 content possible in the products of combustion in the above example with *no excess air* is arrived at as follows:

$C + O_2 \rightarrow CO_2$	·8 lbs. × (2·66 lbs. O_2 + 1 lb. C) =	2·93 lbs. CO_2
$H_2 + O \rightarrow H_2O$	·04 lbs. × $\frac{(2 \text{ lbs. } H_2 + 16 \text{ lbs. } O_2)}{2}$ =	·36 lbs. H_2O
Total products as above		= 11·61 lbs.
∴ $N_2 + H_2O$		= 8·68 lbs.
Wt. N_2	8·68 − ·36	= 8·32 lbs. N_2

	By weight	*Molecular Wt.*	*By Vol.*	%
CO_2	2·93 lbs.	÷ 44	·066	18·1
N_2	8·32 lbs.	÷ 28	·297	81·9
			·363	100·0

With *no* excess air the maximum CO_2 is thus 18·1%

With 50% excess air

CO_2	As above	= 2·93 lbs. CO_2
Net Air 10·61 lbs.: 50% = 5·3 lbs.		= 1·23 lbs. O_2
N_2	8·32 + (5·3 − 1·23)	= 12·39 lbs. N_2

	By weight	*Molecular Wt.*	*By Vol.*	%
CO_2	2·93 lbs.	÷ 44	·066	11·3
O_2	1·23 lbs.	÷ 16	·077	13·1
N_2	12·39 lbs.	÷ 28	·443	75·6
			·586	100·0

With 50% excess air the maximum CO_2 is thus 11·3%

Similarly with 100% excess air it will be found the CO_2 is 8·3%

From the above a series of curves may be drawn relating flue-gas temperature and CO_2 percentage to percentage chimney losses for this fuel, as in Fig. 59, and similarly for any fuel.

Heat Balance and Boiler Efficiency—It is now possible to show how a heat balance is struck. For this purpose a test has to be made on the boiler plant to determine the overall percentage efficiency, i.e. $\frac{\text{Heat output} \times 100}{\text{Heat input}}$. The difference between this and 100 per cent. is made up of the various losses, each of which are calculated separately.

The detailed method of carrying out such a boiler efficiency test is too lengthy to describe here. Reference should be made to British Standard Specification No. 845 for large plants, and 878 for small plants. These are both for steam boilers, but the method is the same for hot water, except that the heat output is calculated directly from water flow and tempera-

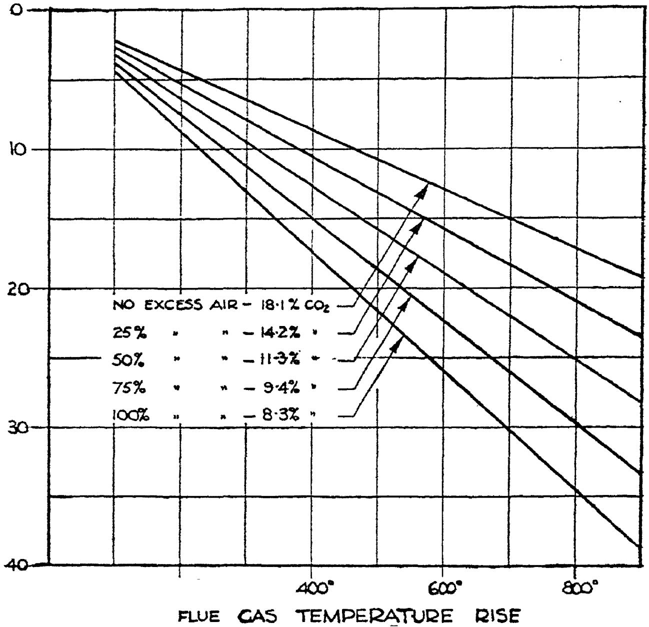


FIG. 59.—Flue Loss for a Given Fuel.

ture rise. Fig. 60 shows diagrammatically the chief factors which have to be measured during such a test, the duration of which is generally 6 hours, with 1 control hour before and after.

The results are then summarized as follows (taking the case stated above with 50 per cent. excess air and assuming a test has been carried out giving an efficiency of 74 per cent. on the gross calorific value and 76 per cent. on the net calorific value):

Heat Account calculated on the Net Calorific Value of the Fuel

Overall thermal efficiency, i.e.: - - - - $\frac{\text{Heat Output}}{\text{Heat Input}} \times 100 =$	76·0%
Loss due to sensible heat in chimney gases (as estimated above) - -	15·62%
Loss due to unburnt CO $\left(\text{assume test showed } \frac{1}{2}\% \text{ CO* } = 59 \times \frac{\cdot 5}{11\cdot 3 + \cdot 5}\right) =$	2·5%
Loss due to combustible matter in ashes and clinker (these are weighed during test and samples taken for estimation of C.V.):	
Say 8% of combustible and C.V. 4500 B.T.U.'s/lb. $= \frac{\cdot 08 \times 4500}{13{,}000}$	2·8%
	96·92%
Radiation and other losses - - - - - - - -	

* B.S.S. 845 gives an empirical formula for estimating per cent. loss due to $\overline{CO}$:

CO

where $k = 59$ for bituminous coal,
61 for anthracite,
68 for coke.

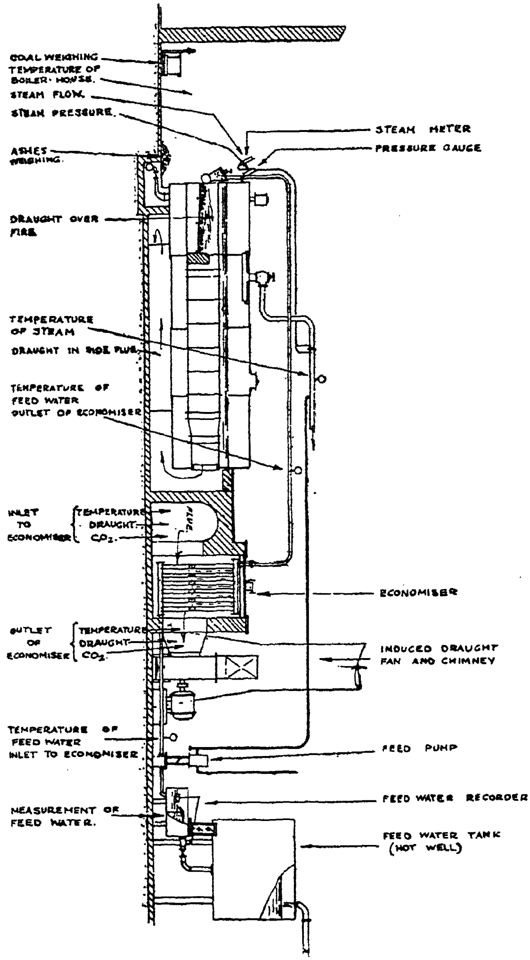


Fig. 60.—Boiler Test Readings.

Fuel and Combustion Notes.

1. *Bituminous coal* is not to be recommended for hand-fired natural draught heating boilers, because of its tendency to smoke and soot up the heating surfaces. It is satisfactory with a forced draught system or automatic stoker.

Anthracite, whilst suitable for hand-firing, may give trouble in automatic stokers, due to the intense heat in the fire pot burning the metal.

2. *Coke* is a highly reactive fuel giving intense radiant heat, is satisfactory for hand-firing, but not for automatic stokers on account of its abrasive qualities. It is best fired in boilers of the magazine type. For

smaller boilers a mixture of coke and anthracite is found to give very good results.

3. *Primary air* is the air supplied under the grate of a boiler promoting primary combustion.

4. *Secondary air* is the air supplied above the fire to supply additional oxygen for CO or unburnt volatiles to be ignited. The benefit of secondary air varies with different fuels, also with the method of firing and depth of bed. Secondary air is more effective if warmed, and there are many so-called 'fuel savers' on the market for attaching to the firing door of a boiler, which do nothing more than provide means for warming the secondary air.

5. *Combustion loss*, as has already been shown, depends chiefly on the excess air and flue-gas temperature. As the excess air increases, so generally does the flue-gas temperature. This at first sight seems to be the opposite of what would be expected, but the phenomenon is evidently due to the boiler surfaces not being able to pick up the additional heat from the gases passing over them at higher speed. Thus, as the excess air increase is due, generally, to too great a draught, the efficiency goes from bad to worse. Careful adjustment of dampers and air inlet door, to give just sufficient air so that the boiler is not starved, is essential.

6. *Air supply to boilers.* It has been shown above how the theoretical air supply for combustion of fuel is determined. Table XIX gives this for various fuels.

TABLE XIX

THEORETICAL AIR REQUIRED PER LB. OF VARIOUS FUELS

Fuel	Theoretical Air Required	
	in Lbs.	in Cu. Ft. at 60° F.
Carbon	11·5	150
Hydrogen	34·5	452
Anthracite coal	11·7	153
Bituminous coal	11·6	151½
Coke	10·0	131
Oil	14·3	187
Gas	10·1	132

In designing any boiler house, provision must be made for this air plus excess air to enter, add, say, 50 to 100 per cent. to the volume stated. If the air enters naturally the free area of opening necessary may be sized at a rate of 500 ft. per minute velocity. Thus, if 1000 lbs. coal per hour is consumed in the boiler, the area of inlet required would be

$$\frac{1000}{500 \times 60} = 10 \text{ sq. ft.}$$

7. *Flame temperature.* The theoretical flame or combustion temperature

may be calculated, given the heat released and the specific heat and analysis of the products. Flame temperatures range between 3000° and 4500° F. for various fuels and gases. Such a figure has little significance in practice, however, since the radiation from flames is not proportional to their temperature. The flame of burning CO gives a more intense radiation, for example, than that of hydrogen, though the latter may be at a higher temperature. Further, due to the close proximity of water-cooled surfaces in a boiler, theoretical flame temperatures are not reached, hence in some systems of burning it is necessary to line the combustion chamber with refractory firebrick, in order to keep the temperature high enough to support combustion. The normal combustion temperature in a boiler furnace is around 2500° F.

RATING OF BOILER HEATING SURFACE

Taking the above-mentioned practical combustion temperature as the highest temperature in the boiler, and with exit flue gases at, say, 400° F., it is possible to draw a curve somewhat as in Fig. 61, showing the fall in temperature as the products of combustion pass through. Below this is given the proportion of heating surface generally provided with a three-pass boiler. In this the primary surface, i.e. that directly exposed to the radiant heat of the fire or flame, often constitutes 50 per cent. of the total, the secondary about 25 per cent., and the tertiary 25 per cent.

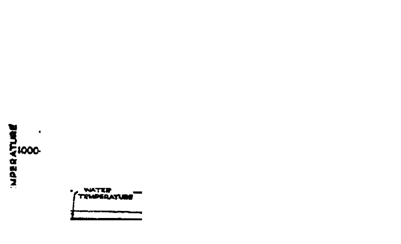


FIG. 61.—Explanatory Diagram showing Temperature Variation through Heating Boiler.

From this it will be apparent that fully two-thirds of the heat absorption of the boiler generally takes place in the primary or direct surface, the secondary and tertiary supplying the remainder between them.

This proportion varies with every type and design of boiler, and the whole subject of heat transfer in boilers is too vast to be dealt with here. Reference to a treatise on heat transmission* is advised for those who wish to embark on a study of this subject. It must suffice here to say that commercial heating boiler designers are now generally more or less in agreement on the following approximate transmission rates:

Primary heating surface	6000 B.T.U.'s per sq. foot.
Secondary heating surface	3000 ,, ,, ,,
Tertiary heating surface	1500 ,, ,, ,,
Average for all surfaces	4400 ,, ,, ,,

* Dr. Margaret Fishenden, *Calculation of Heat Transmission.*

The average figure is for cast-iron sectional boilers of the conventional type. Other boilers with more or less primary surface in proportion to the total have higher or lower average values.

RATING OF GRATE AREA

On the rating of grate area, boiler makers are by no means so much in agreement. Reference to catalogues will show that one type of boiler may have a grate 50 per cent. larger than another with the same output. These variations simply mean that one maker assumes that about 6 lbs. of fuel will be burnt per sq. ft. of grate per hour, whereas another assumes 9 lbs. The boiler rated at 6 lbs. per sq. ft. will require less frequent attention, will not clinker so badly and will be much easier to control, being steadier in its output, but the boiler rated at 9 lb. will be cheaper.

In selecting a boiler it is therefore always wise to check the grate area, and for this purpose Table XX will be found to be of assistance.

TABLE XX

HEATING BOILER RATINGS PER SQ. FT. OF GRATE AREA

Lbs. of Solid Fuel per Sq. Ft. of Grate per Hour	B.T.U.'s per hour per Sq. Ft. of Grate Area, transmitted to Boiler at 70 per cent. Efficiency, using Fuel of Calorific Value:			
	11,000	12,000	13,000	14,000
	B.T.U.'s per Lb.			
4	30,800	33,600	36,400	39,200
5	38,500	42,000	45,500	49,000
6	46,200	50,400	54,600	58,800
7	53,900	58,800	63,700	68,600
8	61,600	67,200	72,800	78,400
9	69,300	75,600	81,900	88,200
10	77,000	84,000	91,000	98,000
11	84,700	92,400	100,100	107,800
12	92,400	100,800	109,200	117,600
15	115,500	126,000	136,500	147,000
20	154,000	168,000	182,000	196,000

For example, a boiler output of 1,000,000 B.T.U.'s per hour (including margin) at 6 lbs. per sq. ft. per hour with fuel 12,000 B.T.U.'s per lb. requires $\frac{1000000}{50450}$ = 20 sq. ft. grate surface, and at 10 lbs. per sq. ft. = 1·2 sq. ft.

For magazine boilers, ratings of 4 to 6 lbs. per sq. ft. of grate are desirable so as to reduce clinkering. Such a rating is, in fact, desirable for any boiler, but it will generally be found to result in an unduly large size with other types, and this may have disadvantages in other directions. Thus for ordinary hand-fired boilers ratings of 6 to 8 lbs. per sq. ft. are normal and practical. Figures of 10 lbs. per sq. ft. and over are high, and should only be worked to for heating boilers when fairly constant attention is available. Steam boilers work at ratings of 20 to 40 lbs. per sq. foot or more as continuous stoking is assumed from the start, but heating boilers are generally required to operate for four or six hours at least without attention, and this calls for a large fire box, possible only with a large grate.

Taking a few typical cases of boilers for various purposes, we see from Table XXI the ratings at which these are catalogued.

TABLE XXI

COMPARISON OF RATINGS OF VARIOUS BOILERS IN COMMON USE

Type of Boiler	Makers' Rating in B.T.U.'s per Hour	Grate Area in Sq. Ft.	Heating Surface in Sq. Ft.	Lbs. Fuel (at 12,000 B.T.U.'s per Lb.) per Hour per Sq. Foot of Grate Area	B.T.U.'s per Hour per Sq. Ft. of Heating Surface
Cast-Iron Sectional (Fig. 51 (*a*))	100,000	1·9	22·6	6½	4430
	250,000	4·2	56·5	7	4430
	500,000	8·5	113	7	4430
	1,000,000	15·9	226	7½	4430
Mild-Steel Sectional (Fig. 54)	500,000	5·1	120	11½	4160
	1,000,000	11·0	240	11	4160
	1,500,000	16·0	340	11	4420
	2,000,000	20·0	448	12	4460
Mild-Steel Magazine (Fig. 55)	1,000,000	24·0	225	5	4450
	1,500,000	34·0	330	5¼	4550
	2,000,000	44·0	440	5½	4550

DRAUGHT

It will be clear that for proper combustion of the fuel it is necessary to have an adequate draught through the boiler. This draught may be

(*a*) natural;
(*b*) induced by a suction fan;
(*c*) forced, with a pressure fan;

but however produced, it will be just balanced by the frictional resistance to the flow of gases through the air door, fuel bed, boiler passages, flue and chimney, plus the kinetic energy of the moving mass of air and gas.

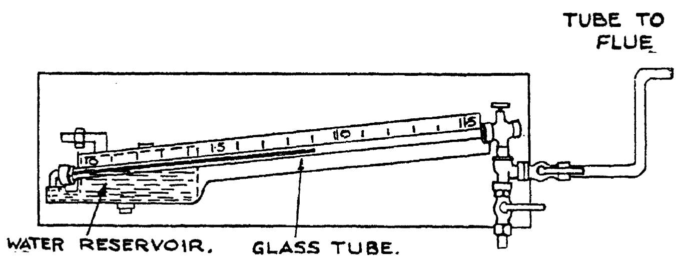


FIG. 62.—Inclined Draught Gauge.

Draught is commonly measured by a draught gauge in inches of water column (see Fig. 62); such a gauge is a common fitment to all large boiler plants.

Natural Draught is produced by the difference in weight between the rising column of hot gases in the flue and the corresponding cold falling column at the temperature of the outside air. Fig. 63 shows the maximum theoretical draught available with various heights of chimney and flue gas temperatures.

Deduction has to be made in practice for loss of head from friction.

The actual draught will generally be about one-half the theoretical.

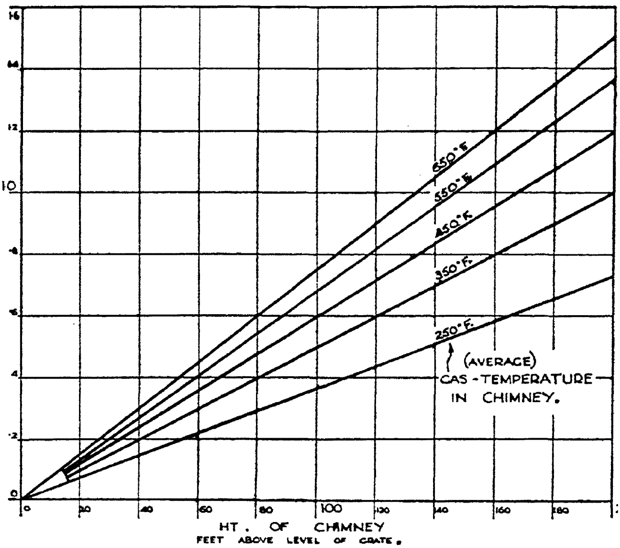


Fig. 63.—Total Theoretical Draught produced in a Chimney (at sea-level).

The limitations of natural draught control the size of fuel permissible, the quality of the fuel, and the rate of burning per sq. ft. of grate area, the resistance to the flow of gases through the boiler, and the boiler efficiency.

The following table gives roughly the rate of burning per sq. ft. of grate area for various draughts, for bituminous coal or coke of C.V. about 12-13,000 B.T.U./lb. under favourable conditions.

Draught at Boiler Dampers: " W.G.	Rate of Burning lbs./sq. ft./hr.
·2	5
·26	10
·33	15
·40	20
·52	25
·67	30
·87	35

Natural draught, it must be remembered, is largely self-balancing.

When, for example, the fire in a boiler is low, the air required for combustion is low, and at the same time the flue gas temperature is low, so that there is little draught to pull this air through the fuel bed. Assume then the air inlet damper to be opened up, the air flow and combustion rate will increase and the flue gas temperature will rise. This latter effect is cumulative since, when the flue gases become hotter, the draught becomes greater, following which the air inlet and combustion rate increase again and the flue gas temperature then rises still further. Eventually a balance is struck when, due to the enhanced temperature of the gases, the boiler heating surface is able to take out sufficient heat to prevent any further rise of flue gas temperature.

A boiler with a good draught will quickly reach this state, often in less than half an hour from a banked state to full duty, and is therefore said to be very responsive. A poor draught such as results from a low chimney serving a large boiler is conducive to sluggish operation, and the state of balance mentioned above may not be reached for two or three hours.

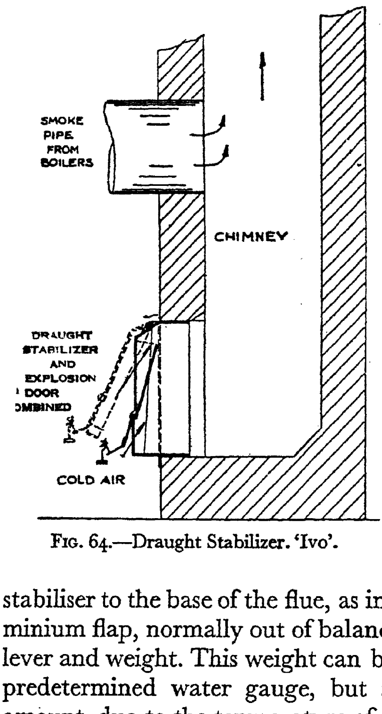


Fig. 64.—Draught Stabilizer. 'Ivo'.

Natural draught is much affected by wind when the chimney is low, and this sometimes accounts for a fire burning out during the night, when the dampers may possibly have been set too wide open. The draught check on the back of the boiler is a useful safeguard through which any excess air called for by the flue due to wind can enter instead of through the boiler.

A better and more uniform control of draught can be achieved at all times by the fitting of a draught stabiliser to the base of the flue, as in Fig. 64. This consists of a hinged aluminium flap, normally out of balance, but kept shut by a small adjustable lever and weight. This weight can be set to keep the flap shut, up to any predetermined water gauge, but as soon as the draught exceeds this amount, due to the temperature of the flue, or to the wind, the flap opens and air is admitted, so 'killing' the draught by the desired amount. This device may be combined with an explosion door as shown.

In some applications, as with oil firing, this fitment is used for the main object of cooling the flue, thus preventing undue warmth on upper floors, which is sometimes a disadvantage.

In Chapter VI the sizing of flues for natural draught is considered, and

curves given from which the best size flue for any given installation can be determined.

Mechanical Draught—With power boilers it has for many years been common practice to install induced draught or forced draught fans, or both.

The *induced draught* fan draws the gases from the boiler and discharges them into the stack as in Fig. 65 (*a*). The fan has to be capable of withstanding high temperatures, and consequently has water-cooled bearings.

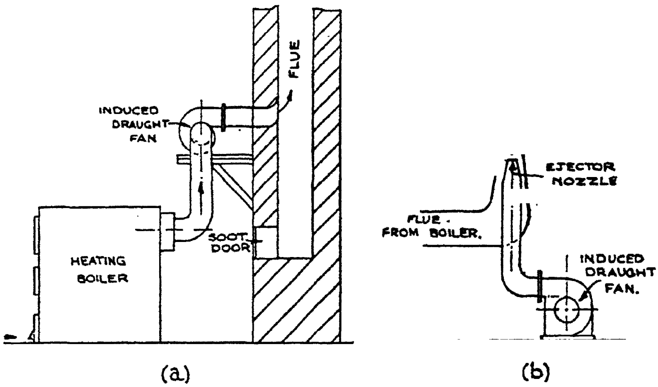


FIG. 65.—Induced Draught Systems.

Alternatively an ejector nozzle in the stack is used to produce the same effect, as in Fig. 65 (*b*).

In either of the above cases the draught over the fuel bed may be up to $1\frac{1}{2}$ in. to 2 in. water gauge, and the air is admitted under the fire naturally.

The *forced draught* system shown in Fig. 66 (p. 126) delivers air under pressure under the fuel bed, or through hollow grate bars.

The positive impingement of the air on to the glowing fuel obviously assists the penetration of the oxygen into the fuel and, as might be expected, greatly increases the rate of combustion for a given grate area.

The combination of forced and induced draught on a boiler or range of boilers provides the most flexible arrangement possible. By this means, a balanced draught is obtainable, whereby the space above the fuel bed is practically at atmospheric pressure, and no excessive air leaks inward or outward are produced.

The application of mechanical draught to small boilers, such as are used for heating, has only come about in recent years. Firing with oil necessitated the use of a forced draught fan for the atomization of the fuel, and it was the advantages accruing from this arrangement which no doubt accounted for the development of the same system for solid fuel, and much attention is being paid to this subject at the present time. It is found that

when forced draught is applied to heating boilers fired with coal, the following benefits are obtained:

(a) A small grade of coal may be used, generally cheaper in price.
(b) Control by thermostat is easily accomplished by the mere switching on and off of the fan.
(c) Quick heating up first thing in the morning is possible, owing to the immediate response of the fire to the air blast.
(d) The fire is kept incandescent, which means absence of unburnt gaseous products, soot and smoke, and a high rate of absorption of heat by the direct heating surface in the boiler.

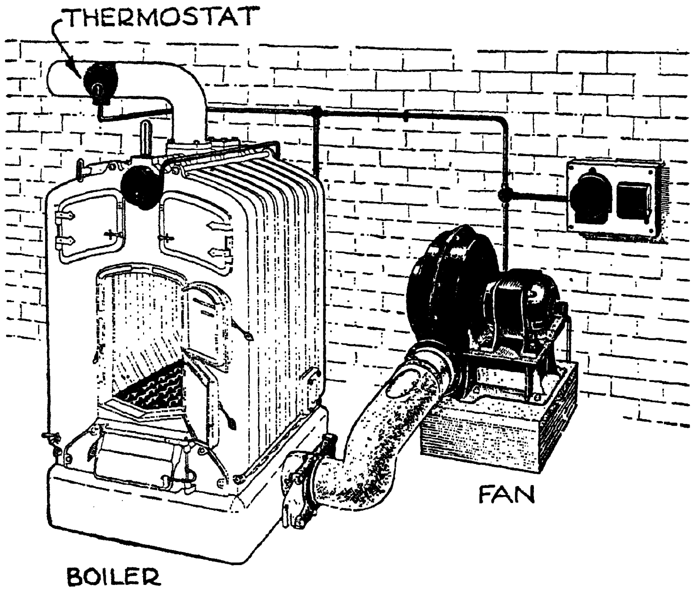


FIG. 66.—Forced Draught applied to a Heating Boiler.

It has been found that economies in running cost may be achieved with such a system, over a natural-draught hand-controlled apparatus.

The disadvantages associated with forced draught are:

(a) Formation of clinker tends to be heavy, and a coal of high ash fusion point is desirable.
(b) Fire bars tend to burn away quickly, due to the blow-pipe effect of the flame on the metal. Often a special alloy is used for the bars to avoid this trouble.
(c) There is a danger of giving a great excess of air to keep a bright fire, and this should be checked with a CO_2 recorder.

Induced draught becomes necessary with any large boiler plant working at high efficiency due to the low stack temperature. Small boilers need

induced draught only when there is some difficulty in providing an adequate flue.

Mechanical Draught and Boiler Design—Much of the energy that is expended in producing improved methods of firing is confined to the application of such systems to existing types of boiler. The fullest advantages cannot be obtained in this way, since boilers designed for natural draught

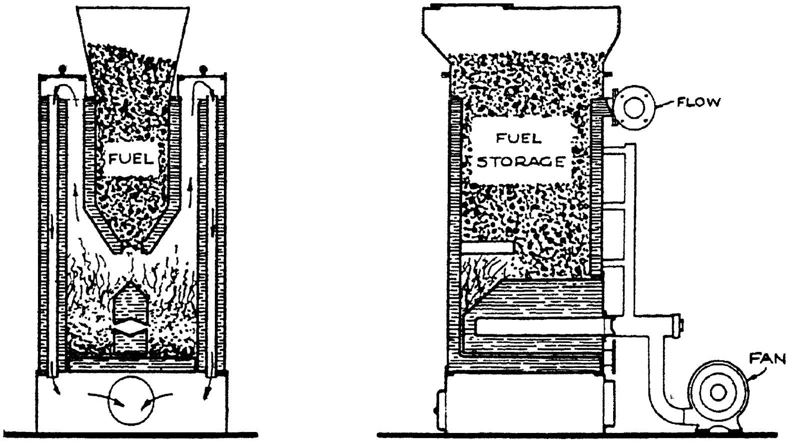


FIG. 67 (*a*).—'Lumby' Forced-draught Boiler. This Boiler incorporates Magazine-feed and has no Grate-bars.

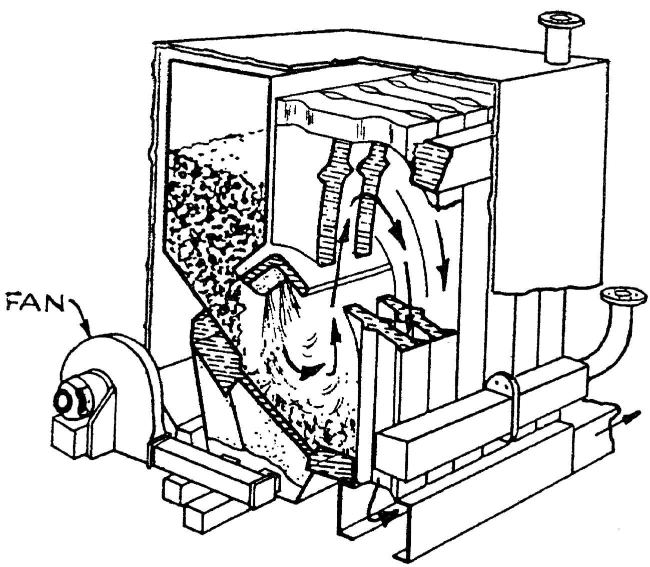


FIG. 67 (*b*).—'Selfstoke' Forced-draught Boiler, using Pre-heated Air, and having a Water-cooled Grate.

have large flue spaces and short gas travel. Given mechanical draught, the flue areas may be much reduced, with consequent increase of gas velocity and rate of heat transfer. Further, the gas travel may be longer, and the exit temperature lower, since the combination does not rely on a high flue temperature to produce draught. Air may be pre-heated so as to increase efficiency still further.

Figs. 67 (*a*) and (*b*) illustrate two types which make use of some of the above-mentioned possibilities, and combine them with a magazine feed to allow of long periods of running without attention.

METHODS OF FEEDING BOILERS

Solid fuel is fed into boilers in one of three ways:

(*a*) by hand;
(*b*) by gravity from a magazine;
(*c*) by automatic stoker.

Hand Feeding—The vagaries of hand feeding have already been mentioned briefly. Hand feeding can be as good as any method if the stoker uses intelligence and forethought. Intelligence in keeping his fire uniform in depth, and incandescent, yet without a great excess of air; forethought in ànticipating changes in heat output necessary at different times during the day and adjusting the controls in advance so as to be producing just the right amount of heat at the right time, thus avoiding waste.

Consider the case of a heating boiler for a house, fired by the maid, or in a small block of offices or flats where it is fired by the caretaker. These people may know nothing of combustion and may even care less; their main object is to keep the inhabitants satisfied with a minimum of effort to themselves. They are therefore liable to err on the side of having plenty of heat by keeping the dampers well open, and a thick bed of fuel heaped nearly up to the crown of the combustion chamber. Under these conditions carbon monoxide is generated in abundance. At the same time the firebox is full and there is no room for the free mixture of air and gases necessary for proper combustion, with the result that the gases are rapidly cooled below their ignition point by the adjacent boiler surfaces, and are carried off to the flue unburnt.

Thus, whilst on test, hand-fired heating boilers may be shown to have efficiencies of 70 per cent. to 75 per cent. under working conditions (during day-time), they may be expected to average about 65 per cent. with good stoking and probably not more than 50 per cent. with bad.

Small fire-pot heating boilers (such as are used in houses), having little or no secondary heating surface at all, on test are stated to be about 70 per cent. efficient, but under working conditions, with unskilled attention, may be as low as 50 per cent.

Gravity Feed—The use of gravity for feeding, as with the magazine type boiler already described (see Fig. 55), is so simple as to require no explana-

tion. The complete absence of moving parts reduces maintenance costs to nil, and attendance, to daily charging of the magazine with coal or coke and the removal of ashes.

The fuel falls on to a sloping grate inclined at the natural angle of repose. Thus, as the lower layers are consumed and drop, more fuel falls down from the magazine, at the same time tending to clear the ash through the bars.

A constant fuel bed depth is therefore maintained, which accounts for the high efficiencies obtainable over considerable periods. Once the dampers and controls have been set, 75 per cent. to 80 per cent. efficiency is obtainable with this type of boiler over long periods.

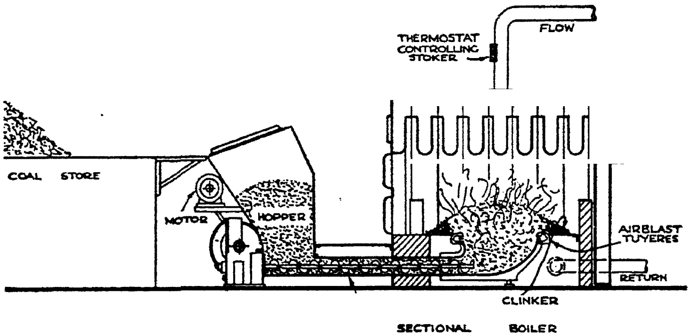


FIG. 68 (*a*).—Typical Installation of Automatic Coal Stoker.

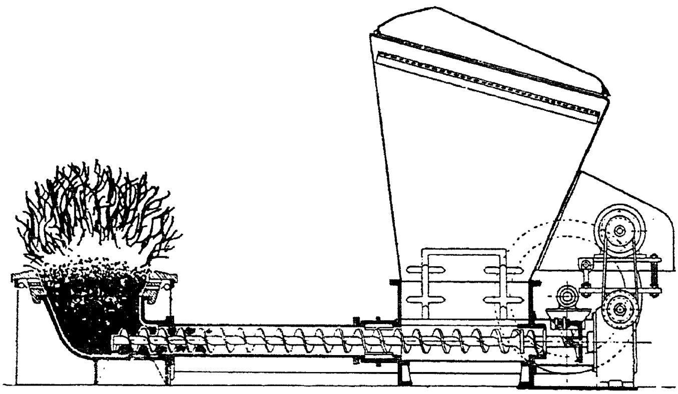


FIG. 68 (*b*).
Typical Automatic-stoker Installation. (*a*) General arrangements.
(*b*) Detail of Worm Feed and Fire Pot.

Automatic Stokers—With one or two exceptions, automatic stokers for heating boilers operate on one principle. Fuel is fed into a hopper, at the bottom of which is situated a worm or screw, rotated at a slow speed

through reduction gearing from a motor drive. The worm is enclosed in a tube beyond the hopper, and serves to convey the fuel into the firepot, which is built into fire brick inside the boiler (see Fig. 68). The fuel must be coal or anthracite of small size, and not coke, which causes excessive abrasion of the worm.

A second tube delivers air into the firepot from a fan driven from the same motor that operates the worm. This air is discharged through a series of slots or openings in the firepot, so disposed as not to be closed up by ash or coal.

Thus, forced draught is provided, and a very high combustion rate is possible, so high, in fact, that a grate is unnecessary. The fuel is burnt as it passes over the edge of the firepot, and all the ash is reduced to clinker in the process. This, however, does not impede the combustion, as the fresh coal brought in by the worm pushes the waste material to one side, requiring periodical removal.

Safeguards are generally provided to prevent jamming of the worm from damaging the mechanism, also to prevent the fire going out if the machine is shut off for lengthy periods by its thermostat. The former is accomplished by a shearing pin, or slipping clutch. The latter, by arranging the motor to start up every hour or so for a few minutes, whether required by the thermostat or not.

Thermostatic control is applied either as stated above, by the stop and start method, or by a variation in the rate of the fuel feed. The latter method should, of course, vary the air supply at the same time, or considerable excess will result when operating at low outputs.

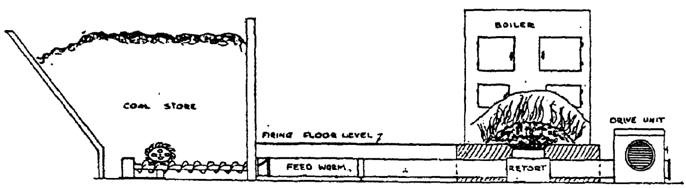


FIG. 69.—Automatic Stoker, Bunker-to-Boiler Type.

Another method of feeding is direct from bunker, as with the type shown in Fig. 69.

The efficiency of a suitable boiler, fitted with automatic stoker, is stated to be 70 per cent. to 75 per cent., and this should be obtainable continuously even with a cheap fuel, subject to clinker removal. 65 per cent. to 70 per cent. is probably more usual in practice.

The advantages of automatic stokers may be summarized as follows:

(*a*) Use cheap small coal.
(*b*) Freedom from the dirt associated with hand firing (this depends on arrangement).

(*c*) Labour costs reduced, as the attendant can carry out other duties during the day.
(*d*) Higher average efficiency if properly adjusted.
(*e*) Absence of smoke.

The disadvantages are cost of current for operation, and maintenance and depreciation cost. These items must be properly evaluated and allowed for in making any comparisons. A further disadvantage when the chimney is close to top-floor windows or to a roof-garden is the fly-ash which is invariably discharged from the flue.

Another form of automatic stoker employing the principle of the producer gas furnace, i.e. the admission of steam with the primary air, has been developed. This type employs gravity feed, and it is capable of burning very low grade fuels such as coke breeze with a high degree of efficiency.

Pulverized fuel, now much used in large boiler plants, has interesting possibilities as applied to heating boilers.

New types of automatic stoker will no doubt continue to be developed to meet changing conditions and fuels.

CONTROL OF BOILERS

Control of output is desirable and necessary in order to limit the heat generated to that necessary to maintain the requisite temperature in the building.

Control may be by

(*a*) hand regulation,
(*b*) simple direct-acting thermostat,
(*c*) indirect thermostat (i.e. electrically operated), controlling primary and secondary air doors, main flue damper and check draught door, or controlling automatic stoker.

Hand control (*a*) need not be further discussed. It is difficult to expect any individual, unless very intelligent, to obtain consistently high efficiency and steadiness of output, with so many variables constantly coming into play, unless the plant is sufficiently big to warrant a complete set of instruments giving him the necessary information.

Simple direct thermostatic control (*b*) has been mentioned before (see Fig. 52), and is useful on small boilers.

Electric thermostat control (*c*) permits a very sensitive thermostat to be used, as it has no work to perform. The temperature-sensitive element is either a bi-metallic strip or a liquid or gaseous expansion bulb making and breaking the circuit by means of contacts or through a mercury tube. The movement of the dampers is effected by means of an electric stalling motor, arranged with reduction gear to move half a revolution in some thirty seconds. The motor is controlled with a limit switch so arranged that having completed one half revolution, it is held with the dampers in the

open position against a spring which serves to return the dampers to the closed position when the thermostat opens circuit, or on current failure.

Primary and secondary air doors, main dampers and draught check door are linked together as shown in Fig. 70, so that the former open and the latter shuts when a rise of temperature is required, and vice versa for a fall of temperature.

Provided the boiler is one that maintains a constant fuel depth, i.e. constant resistance, and that the draught is limited by a stabilizer, it is

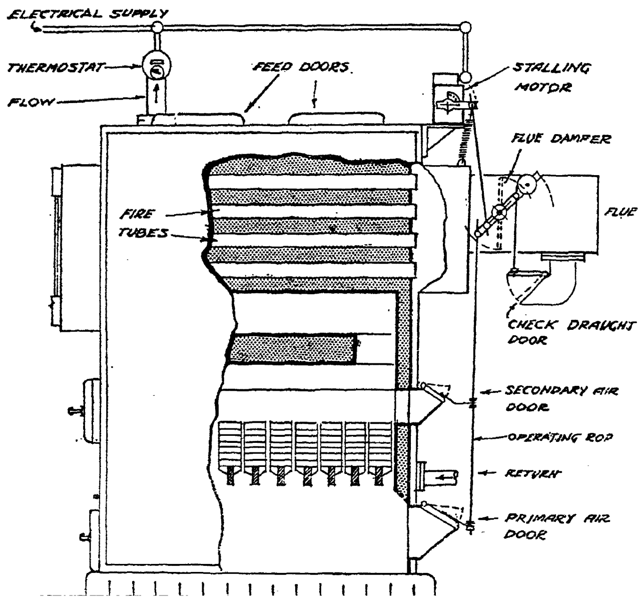


FIG. 70.—Electrical Thermostatic Control applied to Magazine Boiler.

obvious that there must be one position for the various dampers and air doors that gives the highest efficiency. This can be checked by flue gas analysis and temperature and the doors adjusted once and for all at the start. After this the thermostat will cause them to be opened to the same position, and the combustion should remain at a high average efficiency. The thermostat can be operated by water flow temperature (in the case of a water-heating system), steam pressure (with a steam system) or by room temperature, or both.

Control by Weather—Any thermostatic control which gives a constant water temperature for heating suffers from the disadvantage that the temperature setting has to be adjusted for each change of outside weather. The Drayton 'Variostat' is a form of control which has been devised to overcome this disadvantage, operating with two bulbs, one outside and one in the water flow (see Fig. 71 *a*), connected by capillary tubes to a common Bourdon tube all mercury filled. The Bourdon tube rotates a mercury

switch which may be arranged in a circuit to operate a damper motor of the type described above for use with a magazine boiler, or to start and stop an automatic stoker, or again (if the heat is provided by a calorifier) to open and close a steam valve of the motorized type. The two bulbs are

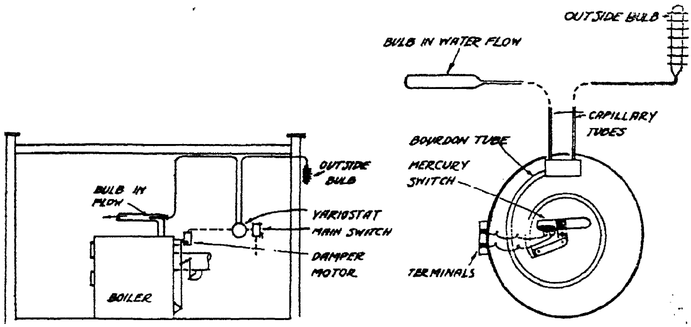


FIG. 71 (*a*).—Diagram of Drayton 'Variostat'.

balanced so that as the external bulb cools the mercury switch closes circuit and starts up the heating agent. The bulb in the flow then becomes gradually warmer until the mercury switch opens circuit and the heat is shut off.

Carrying this principle a step further the Rheostatic Co.'s 'Compensator 4' controller has a small electric heater in the outside bulb, so that it works

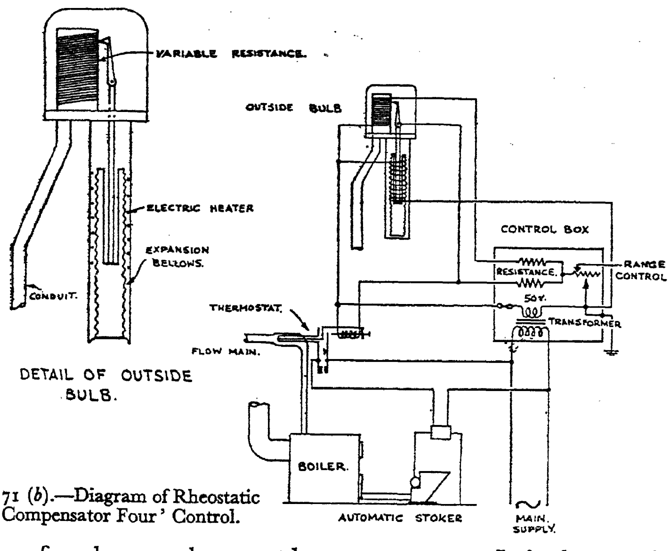


FIG. 71 (*b*).—Diagram of Rheostatic 'Compensator Four' Control.

at some few degrees above outdoor temperature. It is thus sensitive to wind, often an important factor in the heat loss of a building. The device and circuit diagram are shown diagrammatically in Fig. 71 *b*. It is all-electric, so that the outside bulb may be placed at any distance from the

boiler house. Fig. 71 *c* shows a recorder chart from a stoker-fired hot-water heating system controlled by the above. The outside temperature is recorded along with the heating flow, and it will be noticed how as the external temperature rises during the morning and falls later in the day, the flow temperature is brought down and up proportionately. The regular serrations are due to the cutting in and out of the automatic stoker. The operating range and maximum flow temperature are adjustable.

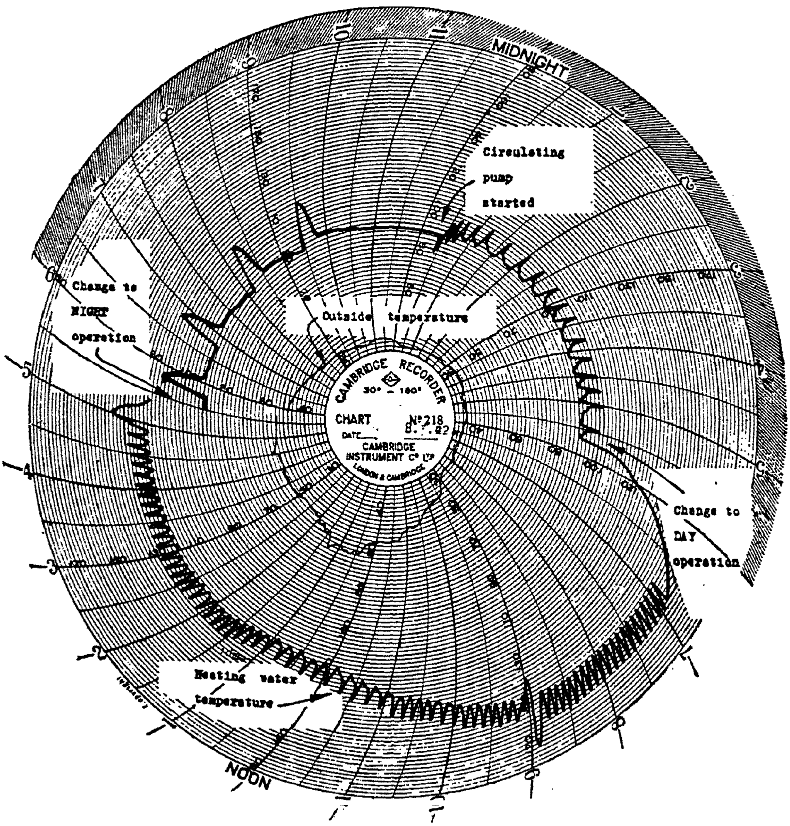


FIG. 71 (*c*).—Temperature Recorder Chart showing Compensator Control.

Another version of this instrument is the Honeywell Brown 'Weatherstat', giving a similar type of control.

For night-time a lower temperature is generally required, and the above controls cater for this by a lower setting controlled either by hand switch or by clock switch.

Complete electrical control with a magazine boiler or automatic stoker of high efficiency might be expected to effect a saving of as much as 25 per cent. of the fuel taken by a hand-fired and hand-controlled boiler.

A word of caution ought to be given about the fitting of complicated thermostatic controls to boilers. Such controls, however well they are made, require maintenance. In the hands of a careful stoker with some mechanical or electrical ingenuity they can be most successful, and the results which the designer intended will be achieved. On the other hand, if they are left untended, dirty, or out of adjustment, they will quickly get out of order and the money spent on them will be wasted.

It is, therefore, to be recommended that, where such controls are fitted, the person put in charge should be something more than a labourer. It is also to be advised that a regular inspection and maintenance contract should be entered into with the firm installing them, so that they are kept in proper order, just as a piano is tuned from time to time.

CLINKERING

Liability to *clinkering* depends chiefly on the temperature of the fire and the fusion temperature of ash, and this in turn depends on its composition. Ash fusion temperatures vary between 1800° and 3000° F. Thus some coals and cokes will clinker at a higher temperature than others, and are more valuable for this reason.

A mixture of small coal and coke breeze will often produce less clinker than if either fuel were used alone. Similarly, a mixture of coals of different composition may be beneficial. This is a subject having a bibliography of its own, which space prevents us from following here.

CONDENSATION IN FLUES

The hydrogen in the fuel produces water vapour when burned, and this may condense into water under certain conditions.

Boilers of high efficiency have a low flue gas temperature at the discharge. If the boiler temperature (i.e. water temperature) is also low (as with panel heating) the dew point of the gases in contact with the boiler plates may be reached, when condensation will occur. This condensate becomes saturated with SO_2 and SO_3 from the flue gases, producing sulphurous acids, which are liable to corrode boiler surfaces and especially boiler tubes. At one important oil-fired installation, bad condensation was experienced at boiler temperatures of 100° to 120° F., but ceased when the temperature was raised to about 160° F. Though water at the lower temperature was required, it was obtained by a mixing device whereby a proportion of hot boiler water (at 180° F.) was mixed with the return water (at 100° F.) to give the desired flow temperature (120° F.).

Anthracite and coke contain less hydrogen than oil, and oil less than gas; hence liability to condensation is in the following order:

Gas (most liable).
Oil.
Bituminous coal.
Anthracite and coke (least liable).

There is also the free water contained in the fuel. This of course depends on whether it has been washed at the pit or delivered wet after standing in the rain. In the case of coke the water used for quenching is probably still contained in the fuel, though the larger gas companies are now using dry quenching. Free moisture may sometimes amount to as much as 10 per cent. of the weight of fuel purchased, an expensive way of buying water.

COSTS OF BOILERS AND AUXILIARIES

Costs of boilers of various types, and automatic stokers, may be taken for approximate purposes from Table XXII.

TABLE XXII

(a) Cost of Boilers, including Fixing*

Makers' Rating in B.T.U.'s/Hour	Type of Boiler			
	Cast-iron Sectional (Fig. 51)	Wrought-iron Sectional (Fig. 54)	Mild Steel Magazine (Fig. 55)	Mild Steel Gravity-feed Type with Forced Draught (Fig. 67 (b))
100,000	£21	—	—	£105
250,000	£37	—	—	£150
500,000	£61	£173	£296	£210
1,000,000	£115	£286	£376	£340
1,500,000	£176	£401	£558	£500
2,000,000	—	£552	£655	£600

(b) Cost of Automatic Coal-Stokers, including Fixing

For Boiler of Capacity:	Cost	For Boiler of Capacity:	Cost
250,000 B.T.U.'s/hour	£132	1,000,000 B.T.U.'s/hour	£234
500,000 „	£186	2,000,000 „	£329

* See note in Preface as to prices.

CHAPTER VI

Boiler Mountings, Chimneys and Heating Calorifiers

BOILER MOUNTINGS (HOT-WATER BOILERS)

Boiler mountings for a hot-water heating boiler comprise:

(*a*) Safety valve.
(*b*) Thermometer.
(*c*) Altitude gauge.
(*d*) Draw-off cock.
(*e*) Damper regulator or thermostatic control equipment.

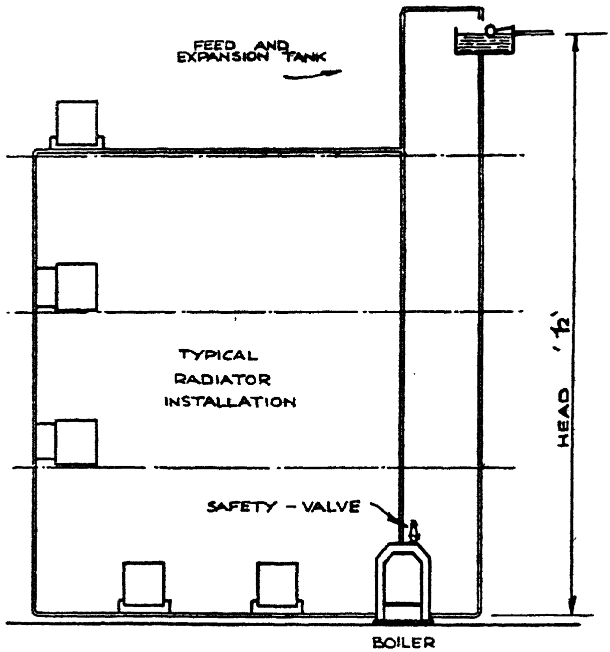


FIG. 72.—Simple Hot-Water Heating Installation.

In addition the following items, though not actually mountings, are necessary accessories:

(*f*) Smoke pipe connection to flue.
(*g*) Open vent pipe.
(*h*) Stoking tools (for hand firing).

(*a*) **Safety Valve**—If we consider the case of a simple hot-water boiler installation, as in Fig. 72 above, we see that the pressure in the boiler is that

due to the head of water '*h*'. Whilst the feed tank remains open to the atmosphere it is clear that this cannot be increased. The fitting of a safety valve would therefore appear to be unnecessary, since the boiler must be strong enough to withstand this pressure with an adequate margin of safety, and, similarly, the safety valve must be loaded to the same head pressure, again with a suitable margin, or it will continually leak.

If, however, the water in the feed tank and vent pipe became frozen and the boiler were lighted, the pressure would rise in the boiler to dangerous limits, and it is to safeguard such an occurrence that a safety valve is required.

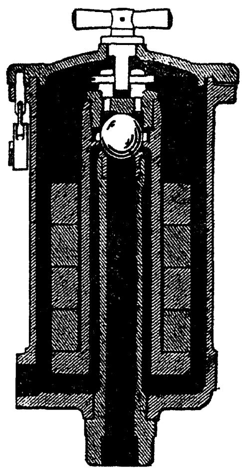


Fig. 73.—Section of Dead-Weight Safety Valve.

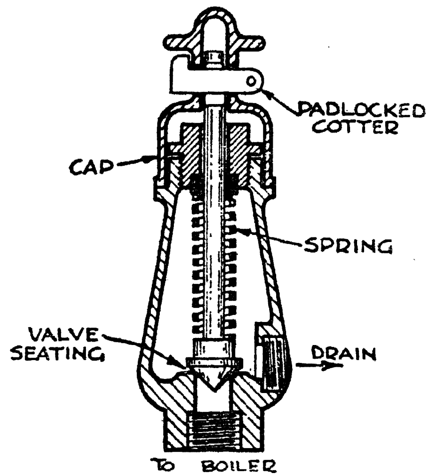


Fig. 74.—Typical Spring-Loaded Safety Valve.

A similar condition would, of course, arise if the feed pipe and vent pipe became choked with fur or scale. Again, when a boiler is fitted with stop valves on flow and return it may at some time happen that the fire is lighted with these shut.

The fitting of a safety valve to hot-water boilers is therefore common practice, and is called for by all boiler insurance companies, though probably less than one valve in a thousand is ever called upon to act, and this is fortunate, as probably few would function if called upon to do so.

The safety valve may be of deadweight pattern as in Fig. 73, or spring-loaded as in Fig. 74. The latter are to be preferred, as they do not suffer from the tendency of the deadweight type to dribble with vibration.

The loading and sizes recommended by the Safety Devices Committee of the Institution of Heating and Ventilating Engineers is given in Table XXIII.

TABLE XXIII

SAFETY VALVE SIZES FOR HOT-WATER HEATING BOILERS

(Recommendations of a Committee of the I.H.V.E. and B.S.S. 779/1938)

(*a*) Loading in lbs./sq. in.

$$= \frac{\text{head of water in feet}}{2} + 10 \text{ (min. 35 lbs./sq. in.)}.$$

(*b*) Rating in B.T.U.'s/hour, based on 4,400 B.T.U.'s/sq. ft. heating surface/hour

(Coal or Coke-fired*)	Safety Valve Size
Up to 900,000	$\frac{3}{4}$ in.
900,000-1,200,000	1 in.
1,200,000-1,500,000	$1\frac{1}{4}$ in.
1,500,000-1,800,000	$1\frac{1}{2}$ in.
1,800,000-2,800,000	2 in.
2,800,000-3,800,000	$2\frac{1}{2}$ in.

* For oil-fired boiler use safety valve one size larger than given in table.

Safety valves, owing to their infrequent operation, tend to become clogged with fur or scale on the seating, and a turn knob is desirable on

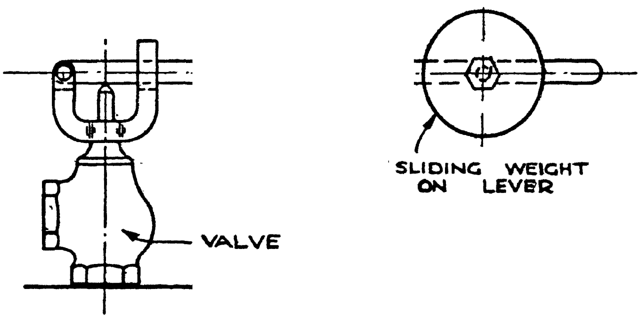


FIG. 75.—Lever-Type Safety Valve (not recommended).

the top so that the valve may be revolved periodically and kept free. A padlock is advisable, to prevent tampering with the adjustment.

Various boiler insurance companies have safety-valve designs of their own, two of which are illustrated in Figs. 73 and 74.

The lever type of safety valve (Fig. 75) is not now in favour, as, amongst other disadvantages, the arm forms a very tempting coat hook. The weight should be locked in position to prevent its being moved.

The outlet from a safety valve should be carried down near the floor with a pipe having a visible discharge.

(*b*) **Thermometer**—The thermometer is a most important accessory to the boiler. It ranges from a plain mercury type in a brass case, costing 5s., to a large dial pattern costing £3 to £4. The latter is desirable where the boiler is large or high.

In order that the thermometer (of whatever type) may be removed for repair, a 'mercury well' is always provided (see Fig. 1 (*b*), p. 4). This is

screwed into the boiler and forms a water-retaining joint. More often than not no mercury is put into these wells, but the only effect of omitting it appears to be to make the instrument slightly more sluggish. A few drops of oil are a good substitute, but this evaporates in time. If mercury is used (as it should be), the cup should not be of brass, as an amalgam will be formed, leading to possible leakage. Steel or special white metal is suitable.

(*c*) **Altitude Gauge**—An altitude gauge is simply a pressure gauge, and is generally marked in feet-head of water, as in Fig. 76. The red index hand is set to the normal pressure in the system, and any variation from this up or down will at once indicate trouble which requires immediate investigation. If the pressure increases, the system is sealed by frost or encrustation, and the fire should be drawn at once. If it decreases, the feed tank is not being replenished and the ball-cock requires attention.

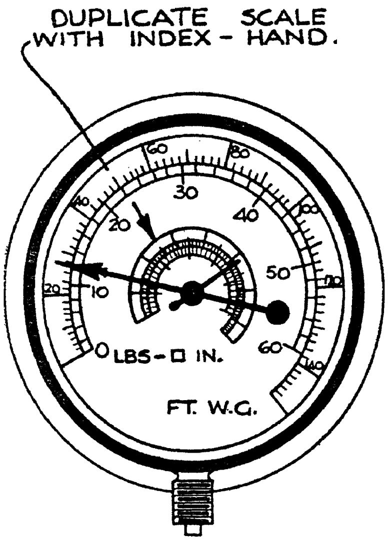


Fig. 76.—Altitude Gauge (graduated in Lbs. per Sq. Inch and Feet Water Gauge).

It should be borne in mind that when a circulating pump is connected in the return, with the cold feed pipe in the suction, the altitude gauge reading will be raised when the pump is running.

It is best to fit a cock so that the gauge can be removed for attention without emptying the system. A syphon of $\frac{1}{4}$ in. or $\frac{3}{8}$ in. pipe is also desirable for connecting the gauge so as to keep it filled with cold air or cold water only. High-temperature water or steam is liable to soften the Bourdon tube, with a consequent error or ageing effect.

Combined thermometers and altitude gauges, as in Fig. 77, are a neat fitment, but suffer from the disadvantage that the gauge portion must always be subjected to the hottest water.

(*d*) **Draw-off Cock**—This item needs little comment, except that a packed gland type of plug cock is the best, as, when opened, a piece of wire can be poked through the opening to free any sludge which may have collected. Being at the bottom of the boiler, this all too frequently is found to be troublesome. A union is desirable for emptying by hose, or alternatively the cock may be coupled with a pipe direct to a sump or drain. In the latter case all bends and tees should consist of crosses plugged at the unused outlets so that mud and scale can be cleared with a wire as already mentioned.

Sizes of emptying cocks are generally:

$\frac{3}{4}$ in. up to about 1,000,000 B.T.U.'s.
1 in. or $1\frac{1}{4}$ in. up to about 2,000,000 B.T.U.'s.
2 in. up to about 4,000,000 B.T.U.'s.
3 in. for sizes over 4,000,000 B.T.U.'s.

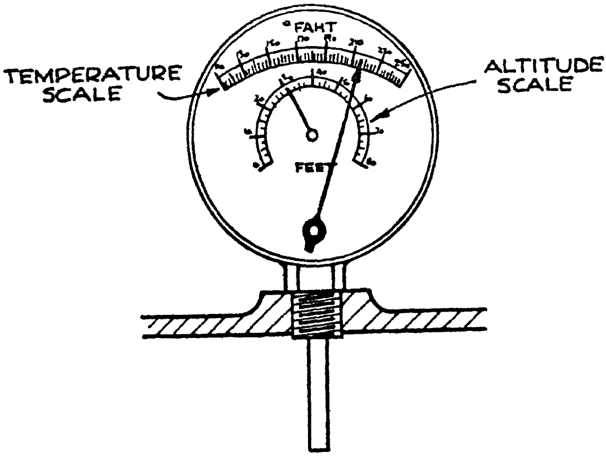


FIG. 77.—Combined Altitude Gauge and Thermometer.

Plate VI (facing p. 152) illustrates mountings *a*, *b*, and *c* in a typical installation. It will be observed from this that the thermometers are of distant reading type, and these, together with the altitude gauges, are mounted on panels over the fronts of the boilers in a very readable position.

(*e*) **Damper Regulator and Thermostatic Control Equipment**—These have already been dealt with at some length, and are only mentioned here to make the list complete.

(*f*) **Smoke-pipe**—From the boiler outlet to the flue a smoke-pipe is necessary. Every endeavour should be made to keep this as short as possible, either by the disposition of the boilers or by the construction of a horizontal brick flue, this being more permanent than a metal pipe, which tends to burn away in winter or corrode in summer with the soot and condensation which collect in it. For the same reasons cast iron is preferable to steel or sheet iron. Standard straight cast-iron pipe and bends in sizes from 4 in. to 14 in. are available, and can generally be fitted in. Above this size and for awkward and difficult runs steel is the only alternative. This is conveniently welded to suit almost any conditions, and should not be less than $\frac{1}{8}$ in. thick, but $\frac{3}{16}$ in. plate is to be preferred, and has a longer life.

Whether of cast iron or steel, all bends should have doors for cleaning and all flues should be swept out at least once in a heating season, preferably at the end.

The authors, in demolishing an old installation which shall be nameless, found a long horizontal smoke pipe, 10 in. diam., of which less than one-third of the sectional area was free, the remainder being filled solid with soot as in Fig. 78.

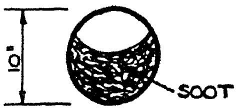


FIG. 78.—Cross-section of a Smoke Pipe (not recommended).

The smoke pipe should not be insulated, as this prevents cooling and adds to the risk of burning away. If the heat from it is a nuisance this can be reduced by enclosing the flue in an outer casing of thinner metal with a 1 in. or $1\frac{1}{2}$ in. air gap between. The lower end of the annular space should be open and the upper be connected to a duct carried outside the boiler house. This allows the free circulation of air over the pipe without excessive heat emission where it is not wanted.

(*g*) **Open Vent Pipe**—The fitting of an open vent pipe direct from the boiler to above the feed tank is the most positive safety device that can be conceived, provided, of course, it is not exposed in such a way as to become frozen in cold weather. It is particularly desirable when stop valves are fitted to a boiler, as is usually the case when two or more boilers are connected together in a 'battery'. Such a vent pipe should be large, as it may in an extreme case have to pass steam, and $1\frac{1}{2}$ in. is the minimum up to 1,000,000 B.T.U.'s per hour, 2 in. or larger for sizes above.

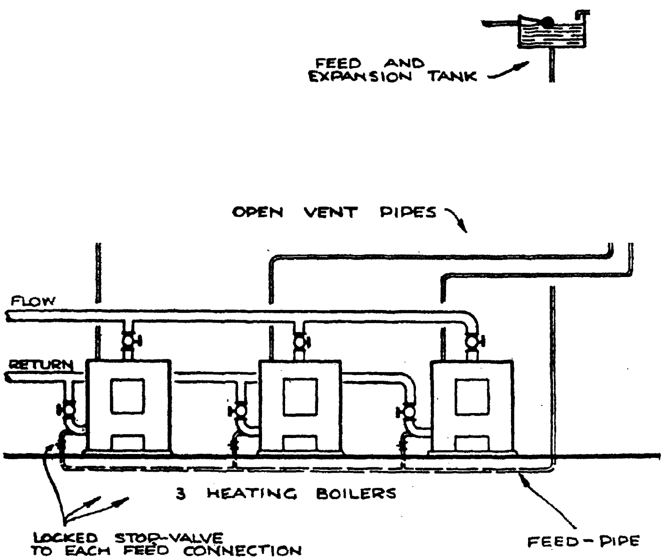


FIG. 79.—Open Vent Pipes for Battery of Heating Boilers.

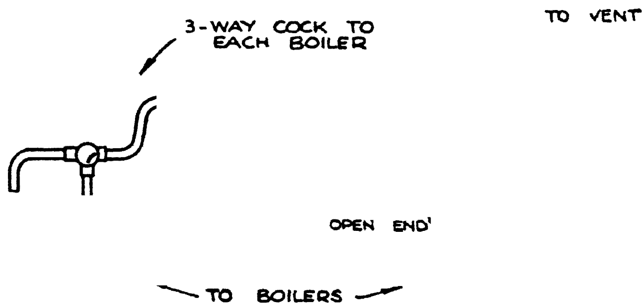
To complete the arrangement, a separate feed pipe to each boiler is really necessary, or, alternatively, a common feed pipe with a stop cock on each branch, locked with a padlock and key.

Such a system would appear as in Fig. 79.

The worst that could happen should both valves on a boiler be shut and the fire alight would be that water and steam would be discharged out of the vent pipe, and this would be replaced by fresh from the tank. If the vents are turned over the tank the same water would be returned, and, apart from noise and hammering, probably no trouble would occur.

In order to avoid the expense of two, three, or more separate vent pipes, all carried to the roof for a corresponding number of boilers, use may be made of a three-way cock as in Fig. 80, one port of each being connected to a single common vent, the second to the boiler, and the third terminates with an open end in the boiler house. Thus each boiler is open to atmosphere either through the vent or through the open end, the latter, of course, only being possible when the flow and return valves are shut. In either case the boiler is relieved from any excessive pressure. This arrangement is not quite as foolproof as Fig. 79.

(*h*) **Stoking Tools**—For solid fuel firing these comprise: shovel, slice bar, scraper, clinker tongs, poker and wire flue brush.



GENERAL ARRANGEMENT AS FIG. 79.

FIG. 80.—Arrangement of Three-Way Cocks on Vent Pipes of Boilers shown in Fig. 79.

BOILER MOUNTINGS (STEAM BOILERS)

For low-pressure steam boilers, such as are used in heating systems, the necessary mountings are as follows:

(*a*) Safety valve. (*b*) Pressure gauge. (*c*) Water level gauges.
(*d*) Blow down cock. (*e*) Automatic boiler feeder.
(*f*) Automatic pressure regulator.

(*a*) **Safety Valve:** the safety devices report already mentioned gives the sizes of safety valves for steam-heating boilers, as in Table XXIV.

TABLE XXIV

SAFETY VALVE SIZES FOR STEAM-HEATING BOILERS

(Recommendations of a Committee of the I.H.V.E. and B.S.S. 779/1938)

Rating in B.T.U./Hour	Min. Clear Bore of Valve (W.P. = 10 lbs./sq. in.)
Up to 80,000	$\frac{3}{4}$ in.
80,000-160,000	1 in.
160,000-260,000	$1\frac{1}{4}$ in.
260,000-400,000	$1\frac{1}{2}$ in.
400,000-700,000	2 in.
700,000-1,100,000	$2\frac{1}{2}$ in.
1,100,000-1,500,000	2 at 2 in.
1,500,000-1,700,000	1-3 in.
1,700,000-2,500,000	2-$2\frac{1}{2}$ in.

Items (b), (c), (d), and (f) do not call for comment.

Item (e) (automatic boiler feeder). This consists of a ball float enclosed in a casing connected top and bottom to the boiler, and so arranged that the ball valve closes the supply when the correct level is reached. Such a device is illustrated in Fig. 195 (p. 337). It is a reliable device which obviates continual attention to the boiler water-level.

Mountings for high pressure steam boilers will not be discussed here, as they are a subject in themselves, and adequate literature on the subject is readily available in mechanical engineers' textbooks.

CHIMNEYS

It has been shown in Chapter V that adequate draught through the boiler is necessary for proper combustion of the fuel, and that with natural draught this is accomplished by choosing a flue which will provide a rising column of hot gas sufficient to overcome the resistances of the boiler, fuel bed, etc.

Size of Flues—The design of chimneys for power plants, etc., is generally done in great detail for each case, having regard to the type of boiler, nature of fuel, material of flue, and so on, but for ordinary heating installations a common rule for flue areas is:

$$A = \frac{\frac{3}{4}g}{\sqrt{h}},$$

where A = cross-sectional area of flue in sq. ft.

g = grate area in sq. ft.

h = height of chimney in feet above grate.

It may be more convenient to proportion the chimney size to B.T.U.'s of maximum rating, and this can be done by assuming a maximum combustion rate of 7 lbs. of fuel per sq. ft. of grate area, and an average calorific value of 12,200 B.T.U./lb. (see Table XX). Then, if R is the rating in B.T.U.'s per hour,

$$A = \frac{R}{80,000\sqrt{h}}$$

Figs. 81 and 82 (p. 146), based on this formula, give the appropriate areas for various boiler ratings and flue heights. The height '*h*' should not be less than 30 ft. and the size never less than 9 in. by 9 in. for the smallest boilers. With oil-fired boilers and forced-draught installations a smaller flue area should suffice than that given by the formula, but it is generally wise to conform to the formula area, as a change-over may be desired later.

Proportion of Flues—Strictly speaking, the curves give the area of the appropriate round flue, and the sizes should be increased by about 25 per cent. for a square flue, since the latter has a relatively greater surface to offer frictional resistance to the flow of gas.

Variations to any considerable extent from the square or circular shape increase this frictional resistance unduly and a rectangle with a ratio of sides exceeding three to one should not be considered. In any case only the area of the ellipse which can be drawn inside a rectangle should be considered as effective area. In a square flue this becomes the inscribed circle.

Excessive Flue Size—It must be remembered that a flue depends for its draught on the difference in temperature between the gases inside and the air outside. A flue much too large is not warmed adequately by the gases and may produce inadequate or spasmodic draught. Hence a flue much in excess of the area given by the formula is as much to be avoided as one that is too small.

This problem is sometimes met when a battery of heating boilers and a domestic boiler are connected to a large common flue. This arrangement may work quite well in winter when all the boilers are on, but be unreliable in summer, when the domestic boiler only is at work on the large flue. Hence it is better to have heating and domestic boilers on separate flues.

This difficulty of giving draught in the summer to the domestic boiler is intensified after a sudden rise of external temperature, when the outer air may actually be warmer than the flue. Under these circumstances, a downward draught bringing fumes back into the boiler house may result.

Flue Construction—As regards construction, chimney flues may either be external or formed as part of a building. In the former case unlined or brick-lined steel with guys is the cheapest for factory work, and self-supporting steel, brick lined, next in order of cheapness. Brick and reinforced concrete are most satisfactory in appearance and permanence.

Reinforced concrete is light, occupies little space and does not require large foundations. Chimneys of this type usually have an outer thickness of 5 to 6 in. of reinforced concrete, an air space of about 2 in., and $4\frac{1}{2}$ in. or 9 in. firebrick lining, which must be free to expand and contract. The lining should always be carried to the top, as when this is not done cracks often occur where the lining ceases.

Plate VII (facing p. 153) shows a typical reinforced concrete chimney serving the boilers shown in Plate IV.

Internal Flues—Chimney flues constructed as part of a building must have a well-insulated lining, both to prevent undue heating of the building and

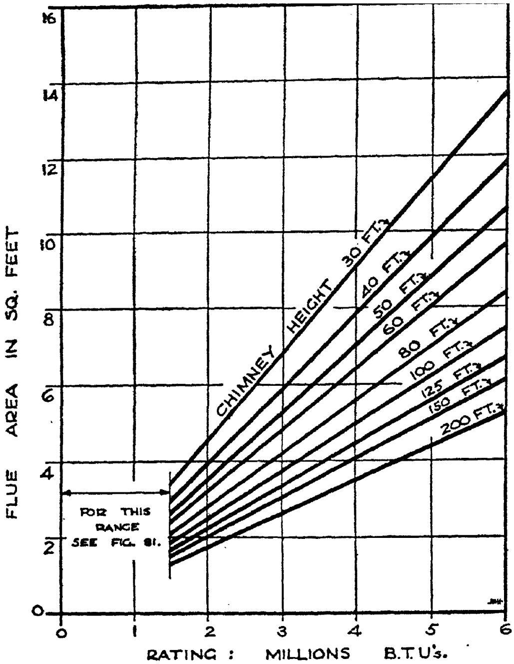


FIG. 81.—Flue Sizes for Ratings up to 6 Million B.T.U.'s.

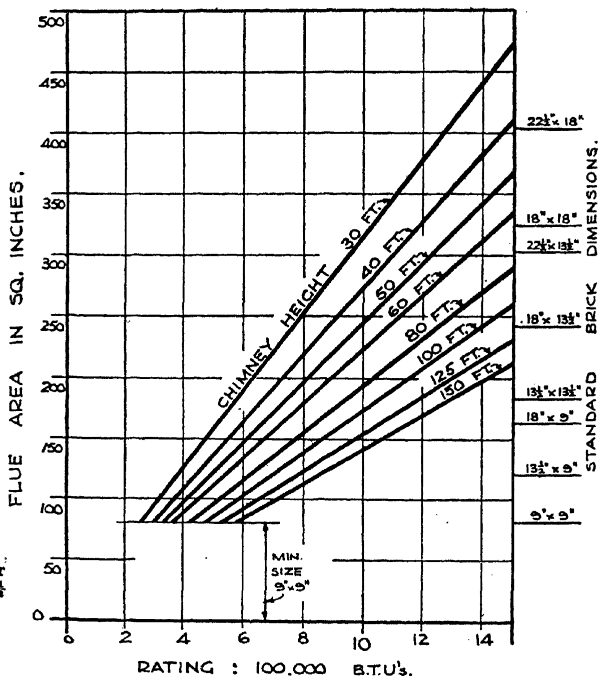


FIG. 82.—Flue Sizes for Ratings up to 1½ Million B.T.U.'s.

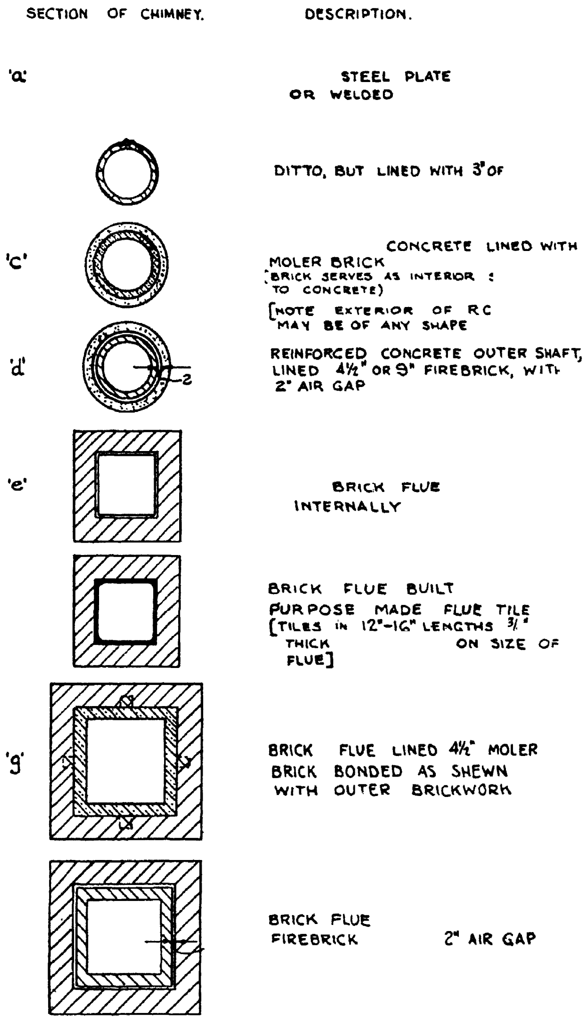


FIG. 83.—Various Methods of Chimney Construction in Order of Cost.

to prevent cracking due to expansion. They should either be lined with firebrick with an air space leaving the firebrick complete freedom for expansion, or with a lining of highly insulating bricks, such as those made from moler or diatomaceous earth, set in mortar containing similar material. These bricks are so highly insulating that they may be bonded with the flue.

Types of flue construction are shown in Fig. 83.

Horizontal Flues should have a flue area not less than that of the chimney, but practical considerations of cleaning, access, etc., often require them to be larger.

An old rule states that the length of a horizontal flue should not exceed one-third of the vertical chimney height, but there are many exceptions to this which function satisfactorily.

Explosion Doors—Explosion doors in horizontal flues are essential with oil and gas fuels and may in certain cases be desirable with coal. Their function is to blow open and relieve the pressure if an explosion occurs.

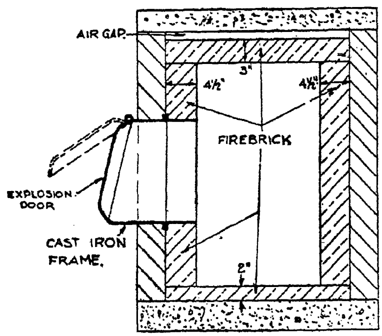


FIG. 84.—Section through Horizontal Flue showing Explosion Door.

This can happen with oil fuel, for example, when the supply ceases for a short time owing to a blockage or a failure of plant, and a few minutes later the supply is resumed. The oil is delivered on to red-hot brick and is gasified. This gas may fill the boiler and flues without becoming ignited until the attendant opens the door, when a violent explosion may occur. A similar result would follow if another boiler connected to the same flue were started up.

Explosions have also resulted from faulty or intermittent electric ignition. Similar risks exist with gas firing, and there are safety devices which go far to eliminate these risks, both with oil and gas, but the provision of explosion doors is still desirable. They can conveniently be combined with ordinary access doors, and they must, of course, open fully, immediately and without restraint.

With coal, and particularly with anthracite, explosions due to CO and volatile gases sometimes occur, and generally take place in the boiler itself at the moment of stoking, slicing, or opening of doors to admit air. Explosion doors are not, however, commonly fitted to flues serving solid fuel boilers.

Fig. 84 illustrates a usual type of explosion door.

INSULATION OF BOILERS

The necessity for boiler insulation has already been referred to. Materials commonly used for the lagging of boilers include plastic fossil meal composition, plastic magnesia finished with a hard-setting material, glass silk and aluminium foil.

Efficiencies and approximate costs for various thicknesses of insulation are given in Table XXV. Fossil meal has a lower insulating value than magnesia, and, owing to the small difference in price, is not to be recommended.

TABLE XXV

BOILER INSULATION

Insulation	Total Thickness	Efficiency, Per Cent.	Approx. Cost per Sq. Ft. Fixed (in London)
Fossil meal plastic composition (not recommended)	1 in. 1½ in. 2 in.	60 65 70	9½d.* 10½d.* 1/-*
Plastic magnesia finished with ½ in. hard-setting material, painted	1 in. 1½ in. 2 in.	80 85 89	10d.* 1/-* 1/2*
Glass-silk, wire netting and hard-setting finish, painted	¾ in. 1 in. 1½ in.	85 89 91	1/8 1/10 2/-
Crinkled aluminium foil. (This must be cased in metal)	1 in.	78	1/6 * Canvas covering 2d. per sq. ft. extra

(See note in Preface as to prices.)

The plastic insulations may be painted direct, but, owing to the surface cracking which invariably occurs, a much better finish is obtained by covering the surface with canvas, pasted on and secured with metal bands. This is painted as required.

Plastic covering should generally be put on in three layers, and a reinforcement of wire netting on top of the first layer is necessary to act as a binding over large surfaces.

Glass silk is frequently secured with a wire netting binding. Over this is placed stout paper, followed by canvas, a skimming of plaster, then paint as before. Alternatively, the plaster may be placed direct on the wire netting.

All coverings to boilers become damaged in time and portions fall off as a result. Thus metallic casings enclosing the boiler and lagging are to be preferred, since they give protection and at the same time present a clean and permanent finish. Several makers have standardized these, finished in vitreous enamel for the smaller boilers and galvanized steel for the larger. Sometimes they are left unlined, but are then of doubtful efficiency.

For a really high finish the metallic casing may take the form of polished aluminium. The low radiating properties of the polished surface in itself assists materially in improving the efficiency of the insulation. Another method is with planished steel as in Plate IV.

It should be remembered that when pipes are insulated their heat transmitting surface is increased due to the greater resulting diameter, and there is in consequence an optimum thickness beyond which the heat emission may actually be increased (this is discussed in Chap. IX). In the case of boilers this does not apply, since surfaces are large and flat, so that the thicker the insulation the greater the heat saved. For hot-water boilers,

however, thicknesses need not generally be more than 2 in. of an efficient material.

HEATING CALORIFIERS

A calorifier is, in effect, a heat exchanger; that is to say, a device whereby heat from a medium at a high temperature is transmitted to a second medium at a lower temperature. The high-temperature medium may be

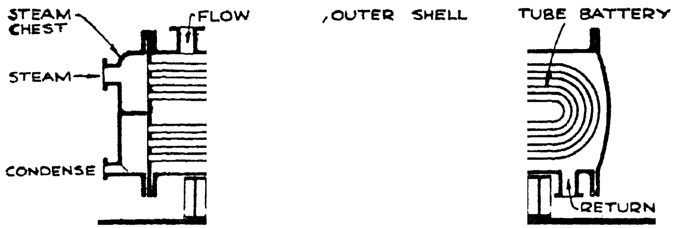


Fig. 85.—Steam Calorifier for Heating.

either steam or hot water, and the low-temperature medium water or other liquid.

The objects and application of calorifiers to heating have already been described in Chap. IV, and it only remains to discuss their design, characteristics and construction.

Types of Calorifier—A horizontal steam calorifier for heating is shown in Fig. 85 and consists of:

(*a*) An outer shell.
(*b*) An internal battery of piping (alternatively radiators are occasionally used).
(*c*) A chamber in which the ends of the tubes terminate so as to admit the heating medium.

The high-temperature medium is generally passed through the tubes, the medium to be heated being outside. This gives a lower temperature on the outer casing and consequently less heat loss by radiation.

The outer shell is either of cast iron or steel, the latter being welded or riveted. More costly metals are generally unnecessary for heating systems in which corrosion troubles are not commonly encountered, though copper is often needed on H.W.S. systems where the same water is not re-circulated. Inlet and outlet connections or flanges are, of course, necessary on the casing. The thickness of shell depends on the head pressure and on the diameter of the vessel; a minimum test pressure should be 80 lbs. per sq. in., increased for higher working pressures, but never less than double the working head.

The tube battery may be of steel, brass or copper. Usually the latter is to be preferred, as it may be thinner than steel, allowing a more compact arrangement of the tubes, apart from which it has a slight advantage in conductivity. Of whatever material, solid drawn tubes are advisable.

The tube battery may be made removable, as shown in Fig. 85, with the tubes of U (or hairpin) form, or a 'floating header' may be adopted, the tubes then being straight (as in Fig. 86). An alternative construction is as shown in Fig. 87, in which the tube battery is not removable without complete dismantling.

As scale deposition does not occur in a heating system in which the same water is constantly re-circulated, there is no real need for the battery

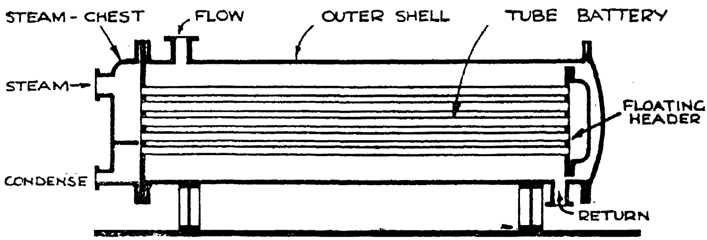


FIG. 86.—Steam Calorifier with Floating Header.

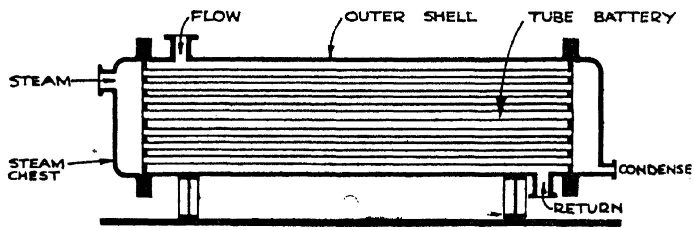


FIG. 87.—Steam Calorifier with Fixed Tube-Plates.

to be readily removable, and the third alternative, which is generally the cheapest, is therefore often adopted.

Those responsible for the maintenance of such plant, however, more usually favour a removable battery, since a defective tube can so much more easily be inspected or replaced.

The chamber for the admission of the heating medium is called the steam chest and is generally of cast iron with inlet and outlet connections for the steam and condense (or flow and return if high temperature water is used). The 'through' type of calorifier (Fig. 87) has, of course, two such chests, one at each end, but the normal U tube and floating header types have one only, with a division to separate the inlet and outlet sides.

The size of the steam chest, or its projection from the front of the tube plate, is dependent on the pressure of steam employed. Exhaust steam entering at atmospheric pressure or below has a comparatively great volume and requires a large entering pipe, hence a large steam chest, perhaps 6 in. or 8 in. from the plate. High-pressure steam may be catered for in a much smaller chest having possibly no more than 2 in. clearance from the tube front.

The steam chest, tube plate and shell are generally machined to the same diameter outside and bolted together with bolts passing through all three, suitable gaskets maintaining the steam and water joints between them. Sometimes the end of the shell is similarly provided with a bolted end for convenience of inspection.

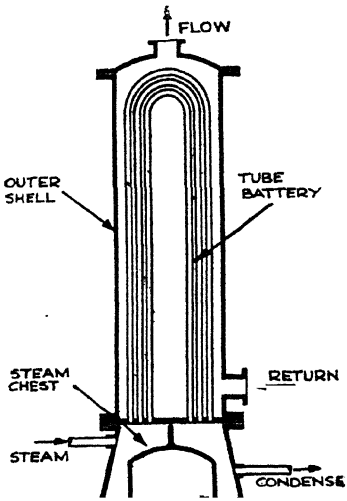


Fig. 88.
Vertical Steam Calorifier for Heating.

Calorifiers may be either horizontal, as illustrated in the figure given above, or vertical, as in Fig. 88. The latter type has its steam chest formed in the pedestal, and the casing is lifted bodily off for inspection. This arrangement occupies less floor space than the horizontal type, and is sometimes more convenient on this account. In either case it will be found that considerable outputs may be catered for in a very small space.

Other considerations apart, long or high calorifiers of small diameter are cheaper than short and wide ones, since the main cost is in the number of tubes and not in their length. The only point to be watched here is that the number and internal cross-sectional area of the tubes is not so small as to cause a large pressure drop for the steam in its passage through. For full effectiveness throughout the whole length of the tube battery full pressure should obviously be maintained to the end, and for this reason the pressure drop should be negligible.

Water inlet and outlets on the shell should be so disposed that the main flow completely traverses the battery. A useful arrangement with a horizontal type is that shown in Fig. 89, wherein a mid-feather in the shell

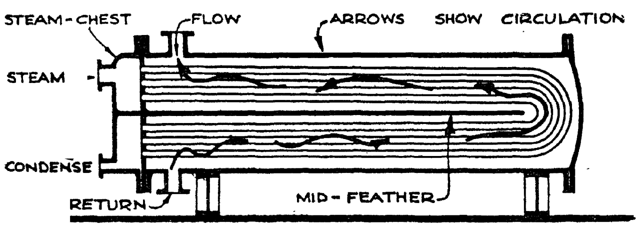


Fig. 89.—Steam Calorifier with Mid-Feather.

diverts the incoming water over the full length of the battery and gives, at the same time, a 'contra flow' effect whereby the coolest water meets the coolest steam and the hottest water the hottest steam, an arrangement which gives the greatest transmission.

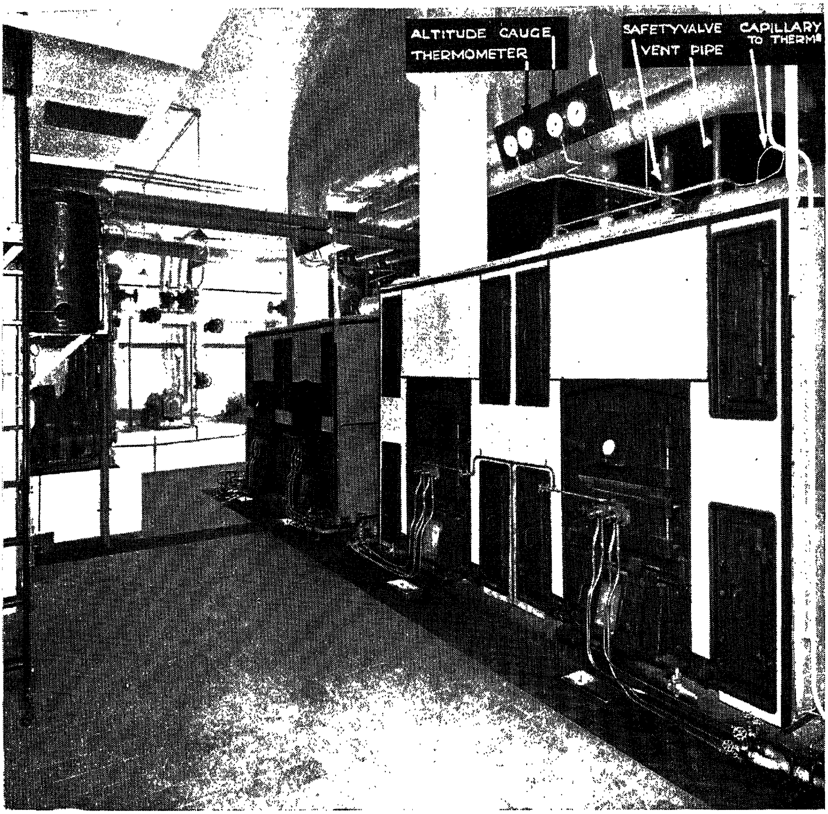


Plate VI. Typical oil-fired boiler installation showing various instruments mounted on panels installed by Messrs Rosser & Russell Ltd. (*see p. 141*)

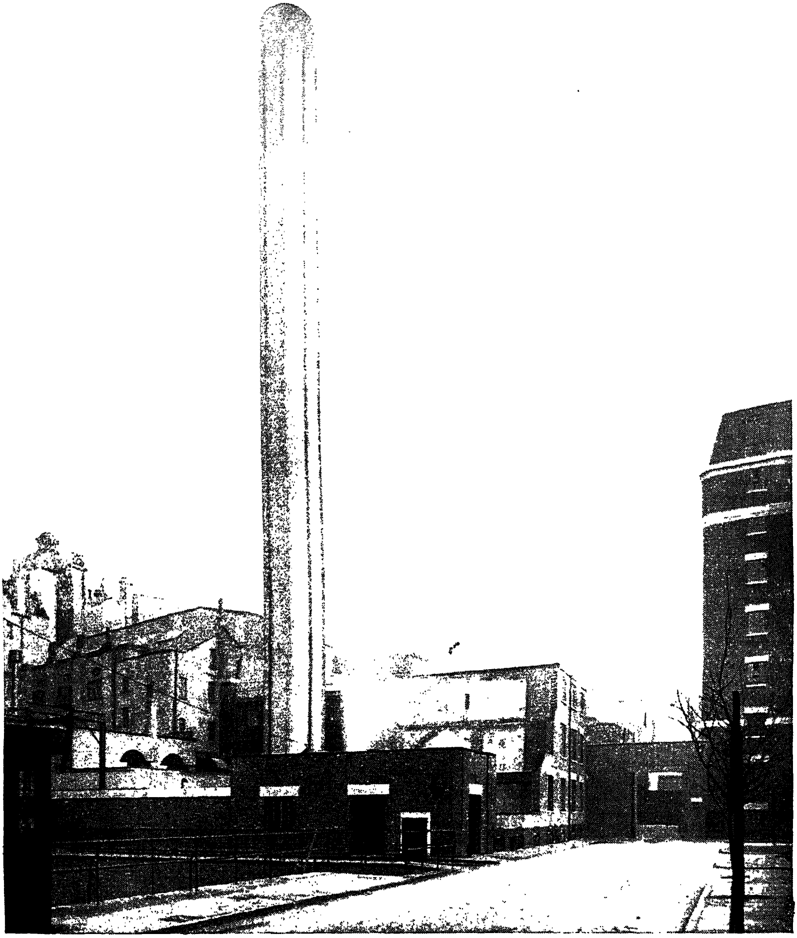


Plate VII. A reinforced concrete chimney to a modern heating installation; St. Luke's Printing Works *(see p. 145)*

Calorifier Tubes—Tubes may be plain, or indented as in Fig. 90. These latter give a larger surface in a smaller space and have greater freedom for expansion than plain tubes with fixed ends, but have no advantage in this respect over U or hairpin tubes. The plate in which the tubes terminate is either of steel or bronze and is called the tube plate, this generally being

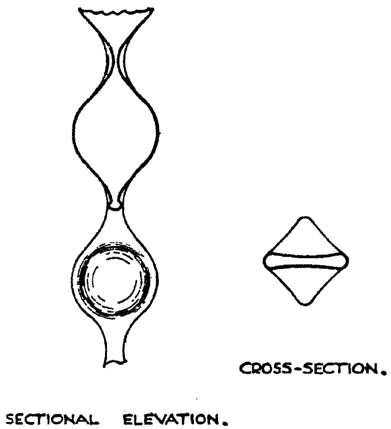


FIG. 90.—Indented Tube for Heating Calorifier.

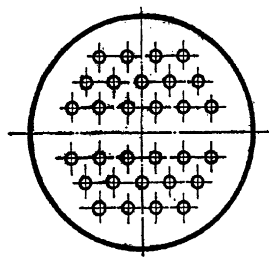


FIG. 91.—'Staggered' Tubes.

about 1 in. in thickness. Plain copper and brass tubes are generally expanded into the plate with an expanding tool; indented tubes are fixed with a screwed thimble connection; steel tubes are either expanded or welded in.

The tubes are commonly arranged in a staggered formation as in Fig. 91, as this leaves a greater amount of metal in the tube plate between the tubes for a given size, and also gives better transmission by causing the rising convection currents to impinge on each pipe with less shielding.

Tubes should in any case be arranged so as to drain freely all the condensation that collects. Hairpins (see Fig. 92) are better than U tubes on

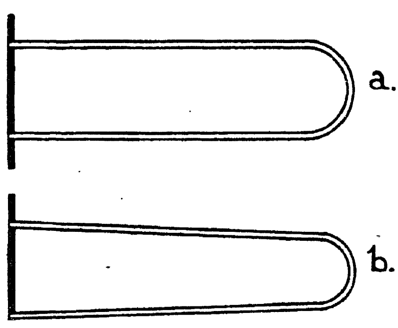


FIG. 92.—Calorifier Tubes: (*a*) U Tube; (*b*) Hairpin Tube.

this account, as the latter often tend to sag at the bottom unless adequately supported. The vertical type is superior on account of its freedom of draining when of the 'through' type, but not in the case of the inverted U form, in which steam and condense flow in opposite directions on the inlet side.

In preparing the layout of the plant, provision of space for tube withdrawal should be borne in mind.

Rating of Calorifiers—The proportioning of the tube surface of the calorifier to the output required is an essential item of calculation.

The transmission rate per unit area of heating surface will depend on:

(*a*) The velocity of the water. This affects the transmission rate considerably if the calorifier is designed on a high-velocity principle. Normally, however, heating calorifiers are not so designed, and consequently no account is taken of the velocity, the assumption being that the full transmission should be possible with only natural circulation.

(*b*) The material of the tubes. This need not be considered as a variable in practice, since the difference in transmission coefficient between steel and copper, which are the two chief materials used, is not more than 3 or 4 per cent., copper being the higher.

The condition of the surfaces of the tubes also has a bearing on the coefficient, as might be expected by analogy with the air-to-air transmission coefficients already discussed.

(*c*) The condition of the steam. It has been found that the amount of air in the steam affects the coefficient, which is lowered considerably by the presence of a small quantity of air.

(*d*) The difference of temperature between the steam and water (or high-temperature and low-temperature water) which must be computed for each case. The temperature of steam depends on its pressure (unless super-heated), and may be taken from any steam table or from Table XXVI.

TABLE XXVI

TEMPERATURES OF SATURATED STEAM

Steam Pressure in Lb./Sq. In. (Gauge)	Temperature in ° Fahr.
0	212°
5	227°
10	239°
15	250°
20	259°
30	274°
40	287°
50	298°
75	320°
100	338°
125	353°
150	366°

The water temperature is the equivalent mean inlet and outlet temperatures. This, strictly speaking, is not the arithmetic mean, but the latter is sufficiently accurate for all practical purposes.

The transmission per degree difference is not constant, but varies with the steam temperature, being greater at higher temperatures.

With the many variables, it is not surprising that the coefficients assumed in practice vary over a wide range, so much so that an enquiry sent to six different makers for a calorifier of given duty will generally result in as many different areas of heating surface being quoted for.

Table XXVII gives some coefficients which have been found safe in practice for steam-water calorifiers, and for water-water where used.

TABLE XXVII

TRANSMISSION COEFFICIENTS FOR CALORIFIERS

(a) *Steam-Water*

Temperature Difference in ° F.	Transmission in B.T.U.'s per Sq. Ft. per 1° F. diff. per Hour
50°	140
60°	160
70°	180
80°	200
100° and over	250

Coefficients given are for heating systems with natural circulation.

(b) *Low Pressure Water-Water*

Coefficient for ordinary temperature differences, B.T.U.'s per sq. ft. per 1° F. diff. per hour: water vel. 1½″/sec.—40, 6″/sec.—70, 12″/sec.—90.

Indented tube surfaces are stated to have higher transmission rates than plain ones, but data on this point are not readily available.

The heating surface required must be computed from the inside diameter of the tube, since this is the side exposed to the high-temperature medium. Table XXVIII gives the length of pipe required per square foot of heating surface for various sizes of commercial tube for copper and steel.

TABLE XXVIII

HEATING SURFACE OF CALORIFIER TUBES

Nominal Diameter of Tube	No. of Feet Run required to give 1 Sq. Ft. of Heating Surface on Inside Surface	
	Iron Tubes (Sized on Internal Diameter)	Copper Tubes (Sized on External Diameter—16-Gauge Copper)
¾ in.	5·1	6·1
1 in.	3·8	4·4
1¼ in.	3·0	3·4
1½ in.	2·5	2·8
2 in.	1·9	2·0

Example of Calorifier Sizing

Output required (including margin, etc., as for boilers)	= 1,000,000 B.T.U.'s/hr.
Steam pressure 20 lbs./sq. in.	
∴ steam temperature (from Table XXVI)	= 259° F.
Required water temperature (flow) = 180° ; ,, ,, ,, (return) = 150° } Mean	= 165° F.
Difference	94° F.
Coefficient per sq. ft. per degree difference (from Table XXVII)	= 200
∴ B.T.U.'s per sq. ft. = 200 × 94	= 18,800
∴ Heating surface required = $\frac{1,000,000}{18,800}$	= 53 sq. ft.
If 1¼" (outside diameter) copper tubing were used (with steam inside the tubes) length necessary to provide 53 sq. ft. (see Table XXVIII) would be 53 × 3·4	= 180 ft. run.

Mountings for Calorifiers are much the same as for boilers. Safety valve, thermometer, and drain cock are essential on the water side, and a steam pressure gauge and trap on the steam side. An altitude gauge connected to the shell is desirable. Open vents direct from calorifiers are not usual.

Thermostatic Control to calorifiers generally effects large savings in

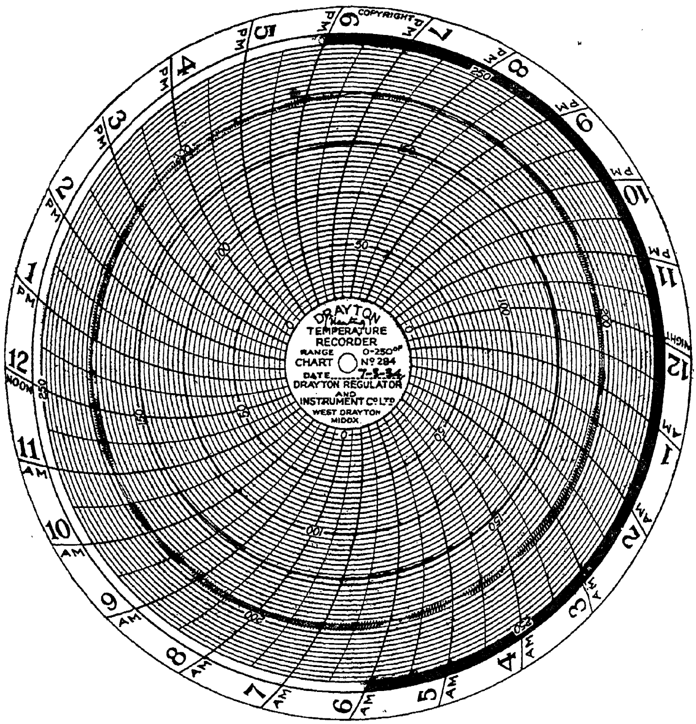


FIG. 93.—24-Hour Temperature Chart showing Flow and Return Temperatures for a Thermostatically-Controlled Calorifier Installation.

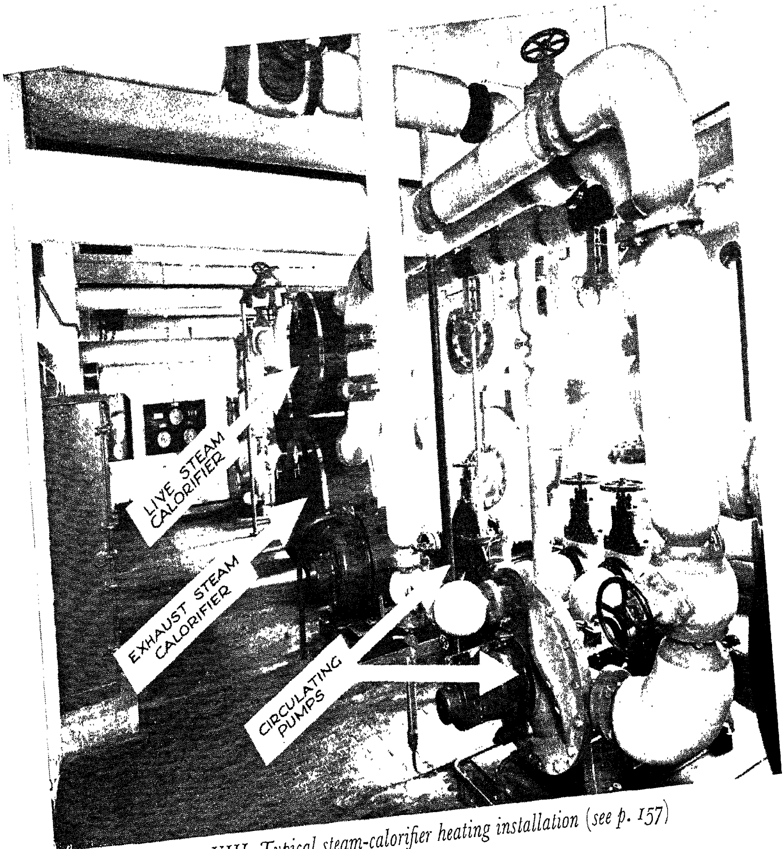


Plate VIII. Typical steam-calorifier heating installation (see p. 157)

Plate. VIIIa. Range of Gravity Feed Coke Boilers (Hartley & Sugden). Installed to the authors' designs by G. N. Haden & Sons at Queen Mary College, London. The bunkers and filling chutes over can be seen in the illustration (*see p.* 109)

steam consumption, and is therefore most desirable. In one installation with which the authors were connected the operation of such thermostats, after the installation was initially started up with no controls in use, effected a saving of 20 per cent. in the metered steam consumption, owing to the elimination of the overheating which had previously taken place. Fig. 93 gives a chart taken from an installation of calorifiers thermostatically controlled, and the very uniform temperature of output will be noticed.

Such control may be operated by direct expansion, but the water-operated relay type is much to be preferred for sensitivity, tightness of shut-off, and length of life. Unfortunately, the latter require maintenance for effective continuous operation. Electric motor-operated steam valves with electric thermostats in the water flow are equally positive and efficient, but suffer from the disadvantage of remaining open if the current should fail when they are in this position. This might in some cases prove dangerous, due to excessive rise of temperature. A solenoid type valve is to be preferred for this reason, this being self-closing on current failure.

General—Other items required in connection with calorifiers are supporting cradles, runways or other means for tube withdrawal, and lagging. These do not call for comment.

Table XXIX gives approximate costs and capacities of calorifiers of the cast-iron shell type with copper tubes.

TABLE XXIX

Approximate Costs and Sizes of Heating Calorifiers

Heating Surface in Sq. Feet	Capacity in B.T.U.'s per Hour, with Water at 150° F. Mean		Overall Size, Inches		Approx.* Cost, including Fixing
	Steam at 5 Lb./Sq. In.	Steam at 50 Lb./Sq. In.	Length	Diameter	£
5	80,000	150,000	40	17	20
7½	120,000	225,000	53	17	22
10	160,000	300,000	50	18	24
12½	200,000	375,000	54	18	27
15	240,000	450,000	58	18	30
20	320,000	600,000	70	18	36
25	400,000	750,000	60	23	40
30	480,000	900,000	65	23	44
35	560,000	1,050,000	72	23	48
40	640,000	1,200,000	80	23	52
50	800,000	1,500,000	88	23	60
60	960,000	1,800,000	80	27	68

* See note in Preface as to costs.

A view of a steam-calorifier installation is given in Plate VIII. The normal heating is done with exhaust steam from generating sets in an adjacent building, but should a heavy demand cause the flow temperature to fall, an automatic control admits live steam from the boilers direct to

the live steam calorifier. The effectiveness of this control is evidenced by the record in Fig. 93, which was obtained from the plant illustrated.

In this case the flash steam from the H.P. traps is passed into the exhaust calorifiers.

The circulating pump normally used is driven by a steam turbine seen in the foreground, an electrically driven pump serving as a standby. The exhaust steam from the turbine is used for heating the water, so that the cost of circulation is almost nil.

CHAPTER VII

Oil Firing

Oil, being ashless, is an ideal fuel for the firing of boilers; being a fluid it is more easily handled than coal or coke and can be stored in positions impossible with the latter fuels. It is, moreover, relatively more dense, and, having also a higher calorific value per unit of weight, does not need to be stored in such large quantities.

Table XXX shows the comparative space occupied by one ton of oil, coal and coke, together with the potential heat value in each case.

TABLE XXX

COMPARATIVE VOLUMES AND HEATING VALUES OF FUELS

Fuel	Lbs. per Cu. Ft.	Cu. Ft. per Ton	Calorific Value in B.T.U.'s per Lb. net	Potential Heat-Value in B.T.U.'s per Cu. Ft. of Fuel
Oil - - -	56	40	18,000	1,000,000
Anthracite -	50	45	14,000	700,000
Coal (average) -	45	50	12,500	563,000
Coke - - -	25	90	12,000	300,000

Further advantages of oil fuel are cleanliness, convenience and flexibility of operation, and saving of labour generally. The former have a money value which is difficult to assess, but nevertheless very real.

Its disadvantage is cost, which is higher than that of coal or coke, though many cases exist where this is balanced by a corresponding saving in labour. Similarly, its great facility for automatic control often brings about greater economy of heat than is possible with a hand-fired, hand-controlled system.

Oil is an imported fuel and for this reason is at present being supplanted by creosote-pitch, which is considered at the end of the chapter.

Types of Fuel Oil—Fuel oil as commonly supplied for boilers is the product remaining after the various petroleum spirits have been removed by distillation from crude oil. Owing to the fact that such oils are obtained from wells in various parts of the globe, they vary considerably in their physical and chemical characteristics. All can be burnt in boilers, provided the right type of equipment is provided for the purpose.

Characteristics—Table XXXI gives the characteristics of three typical oils suitable for use in boilers. *A* is a light oil suitable for burners incapable of using heavier oils, *B* is heavier, commonly known as '200 seconds' oil, and *C* is a home-produced oil from the low-temperature carbonization process.

TABLE XXXI

CHARACTERISTICS OF TYPICAL FUEL OILS

	Oil *A* (Shell Domestic)	Oil *B* (Britoleum)	Oil *C* (British)
Viscosity (Redwood No. 1 at 100° F.)	40 sec.	200 sec.	100 sec.
Specific Gravity at 60° F.	0·88	0·92	1·01
Flash point (closed)	165° F.	175° F.	190° F.
Pour point	Below 20° F.	Up to 30° F.	32° F.
Calorific Value in B.T.U./lb.:			
Gross	19,350	18,900	17,000
Net	18,150	17,800	16,400
Carbon	85·5%	85·5%	85·8%
Hydrogen	13·0%	12·0%	7·9%
Sulphur	1·25%	2·0%	·5%
Asphalt, etc.	·25%	·5%	5·8%

The flash point (closed) is the minimum temperature at which the vapour will ignite in a closed vessel and should not be less than 150° F. This temperature is important, since it means that at ordinary temperatures and far above them oil and its vapour are non-inflammable. It is, in fact, difficult to set oil alight by plunging a lighted taper into it, the taper being extinguished.

The viscosity as determined from Redwood's tables is the time in seconds for 1 c.c. to pass through an orifice of specified dimensions. No. 1 table (fine orifice) is for light oils, No. 2 (larger orifice) for heavier oils.

The pour point generally follows the viscosity, and is the temperature at which the oil ceases to run freely. For practical reasons this point must be below the normal temperature of the storage vessel.

The calorific values (gross and net) have already been explained in Chap. V, and need not be referred to again. Petroleum oils vary but little in this respect.

The sulphur content determines the quality of obnoxious gases emitted from the flue and should be as low as possible for obvious reasons.

The asphalt content is not so important when the oil is used in boilers as it is in the case of Diesel engines, in which an excess of this constituent may cause gumming of valves, coating of cylinders, etc.

COMBUSTION OF OIL

If a piece of rag is dipped in fuel oil, the oil may be burnt, though, as stated above, it cannot be ignited when a flame is plunged into it in bulk. When soaked into a rag the oil presents a much greater surface to the flame, and the ignition point is reached without any surrounding oil being present to conduct the heat away.

The flame from the oily rag will, however, be found to be extremely smoky and entirely unsuitable for use in a boiler. This is because the oxygen of the air cannot penetrate the oil vapour quickly enough to reach all the molecules of carbon before they are driven off beyond the zone in which combustion is supported.

In order to burn oil smokelessly it is therefore necessary

(*a*) to ensure that the oil is finely divided or atomized so as to present as great a surface as possible to the atmosphere;

(*b*) to retain around the combustion space a material at high temperature so as to support combustion throughout as great a volume as possible.

The first requirement (*a*) has given rise to three main types of oil burner which will be described later.

The requirement (*b*) is generally met by providing in the combustion chamber of the boiler a mass of fire-brick, which itself becomes incandescent and so maintains a zone of high temperature around the flame. Until this becomes heated, quantities of smoke may be emitted, and this is often noticeable when an oil burner first starts up.

OIL BURNERS

The three main types of oil burner are:

(*a*) Pressure jet. (*b*) Compressed air or steam jet.
(*c*) Low-pressure air.

(A fourth type depending on natural draught atomization is available, but is not in sufficient use to warrant description here.)

(*a*) **Pressure Jet System** (see Fig. 94)—This is the method employed in marine work and is commonly used on land in large boilers of the Lancashire, Economic or water-tube type. Plate IV illustrates a typical installation employing this method of firing.

The system comprises essentially an oil heater, pump, jet, and furnace front. Steam is generally used for the heater and pump and is controlled so as to maintain an oil temperature of about 140° F. and an oil pressure of 40 to 100 lbs. per sq. in. The jet consists of a hardened steel nozzle with a series of holes drilled tangentially to give a swirling motion to the oil, but the detail varies with every make of plant. The jets are calibrated for a definite delivery in gallons per hour and for change of load a change of jet is necessary. Thus the jet assembly is always made removable. The heater is necessary to reduce the viscosity so that the oil may be atomized by pressure alone.

The purpose of the furnace front is to give to the air entering naturally for combustion a swirling motion opposite in direction to that of the oil spray, so bringing about a close intermingling. By adjustment of the intake deflector the amount of air admitted may be varied to suit the flame, or the air may be shut off entirely when the burner is closed down, so preventing cooling of the boiler.

The system is suitable for large boilers only, but for these it is cheaper than other systems. High efficiencies can be obtained, and the space required for the oil-firing plant is small. Any grade of oil can be burned, and

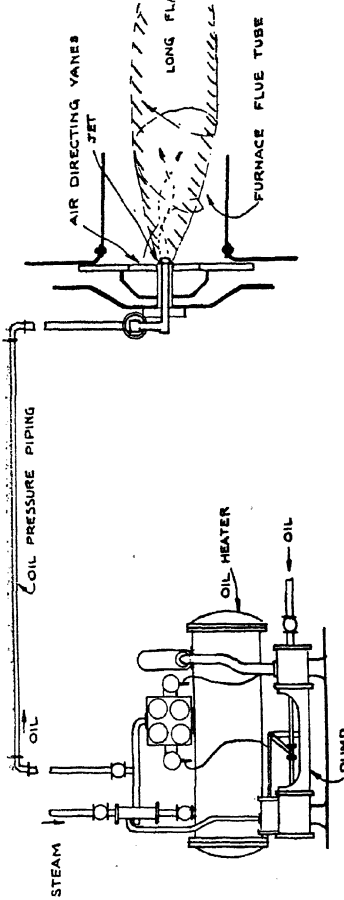


FIG. 94.—Oil Burning, Pressure Jet System.

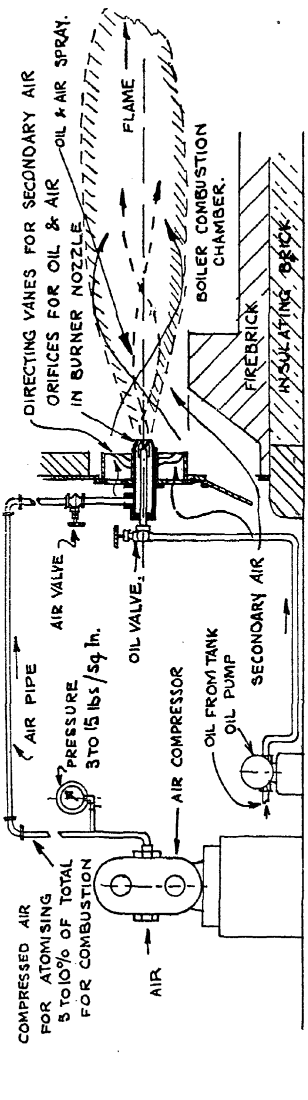


FIG. 95.—Oil Burning, Compressed Air System.

no electrical supply is necessary for operation. A donkey boiler is required for starting. Skilled attendance is required.

The oil piping, being under pressure and at high temperature, requires to be to a special specification as laid down by the Board of Trade, as danger from fire exists should a joint burst.

(*b*) **Compressed Air System**—With this system the atomization of the oil is brought about by a blast of compressed air at the point where the oil is leaving the nozzle, as shown in Fig. 95. In some types the oil enters and is discharged by gravity, and in others by pump under slight pressure.

The air for atomization generally constitutes some 10 per cent. of the total required for combustion and the remainder is allowed to enter through openings in the boiler front or around the burner.

The apparatus comprises an air compressor operating at a pressure of 3 to 15 lbs. per sq. in., an oil pump, the burner nozzle and air-controlling furnace front.

Warming of the oil to about 100° F. is generally advocated with this system to prevent smokiness in the flame, and a heater with steam or hot water coils, with electric immersion heaters for starting, is provided for this purpose. The oil pump is arranged to circulate the warm fuel through a pipe running past each burner and back to the heater so as to leave the smallest possible amount of cold oil in the feed to the jet.

The advantages of this method are low price and high relative efficiency.

The main disadvantage is that it tends to be noisy, but in many cases this is not important.

Plate VI (facing p. 152) illustrates a battery of boilers fired with this system. The compressors will be seen on the left, and each boiler has two burners on account of the size. Gas ignition is provided by the swivel arms seen in the fronts of the boilers.

A variation of this system operates with a steam jet instead of compressed air. This need not be considered in detail here, however, as it has little application in heating practice.

(*c*) **Low-Pressure Air System**—The principle of the low-pressure air system is shown in Fig. 96. Air at low pressure (about $\frac{1}{3}$ lb. per sq. in.) is delivered by a blower into the burner nozzle and is here (in one type) arranged to rotate a cup at high speed. The centre of the cup is fed with oil by gravity or by pump, and the motion of the periphery causes a finely divided spray to be whirled off the edge tangentially. The incoming air is at the same time deflected in the opposite direction by vanes in the nozzle and a most satisfactory flame is produced by this means.

The air volume introduced by the fan is generally about 20 per cent. of the total required for combustion, the remainder entering naturally through apertures provided for the purpose. In some instances, however, the full 100 per cent. of air is delivered by the fan, and this obviously gives more accurate control of the oil-air ratio.

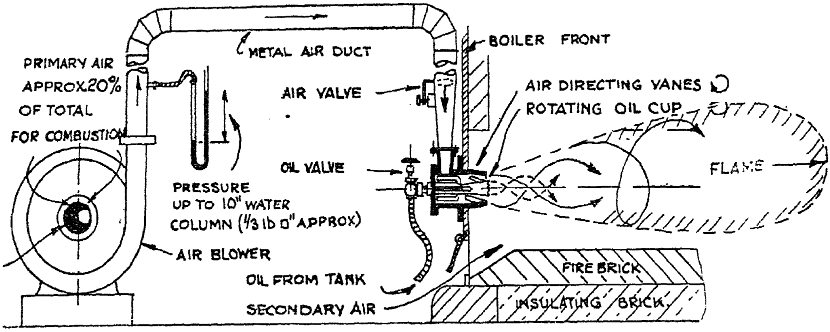


FIG. 96.—Oil Burning, Low-Pressure Air System.

For handling the heavier oils some preheating is necessary, but with light oils this is not essential.

The equipment in its simplest form comprises low-pressure fan and burner only. The air piping has to withstand little pressure and is therefore of light metal construction. The oil has small head pressure and no special piping is required.

The system is probably simpler to operate than others, but, unless proper controls are provided, may be rather lower in efficiency owing to the great ease with which too much primary air may be introduced from the fan in the desire to produce a bright clean flame. This excess air may reach very high values, particularly if the oil supply is cut down below its rated duty.

This system is generally higher in cost than the compressed-air system, but is much quieter in running and therefore often to be preferred. It is applicable to a wide range of duties from 100,000 B.T.U.'s per hour upwards.

Plate XI, the Bank of England boiler house, shows this system as applied to Economic type boilers each of 9,000,000 B.T.U.'s per hour. Plate XA (facing p. 165) illustrates its application to steel sectional boilers of 2,000,000 B.T.U.'s per hour. The low-pressure fans are not shown and the air and oil piping is carried in the trench in the floor covered with chequer plating.

Flame Characteristics—Of the three main types of oil burner, the pressure-jet system has the longest flame, and for this reason a lengthy cylindrical combustion chamber such as is provided in a Lancashire or Economic type boiler is particularly suitable. In these the flame appears to fill the flue completely.

The compressed-air system has a somewhat shorter, more fierce flame, but it still requires a lengthy travel before the gases pass away from the high-temperature zone.

The low-pressure air system produces a soft gaseous flame high in radiant heat value and is of such flexibility that it may be adapted to operate successfully in almost any shape or size of combustion chamber.

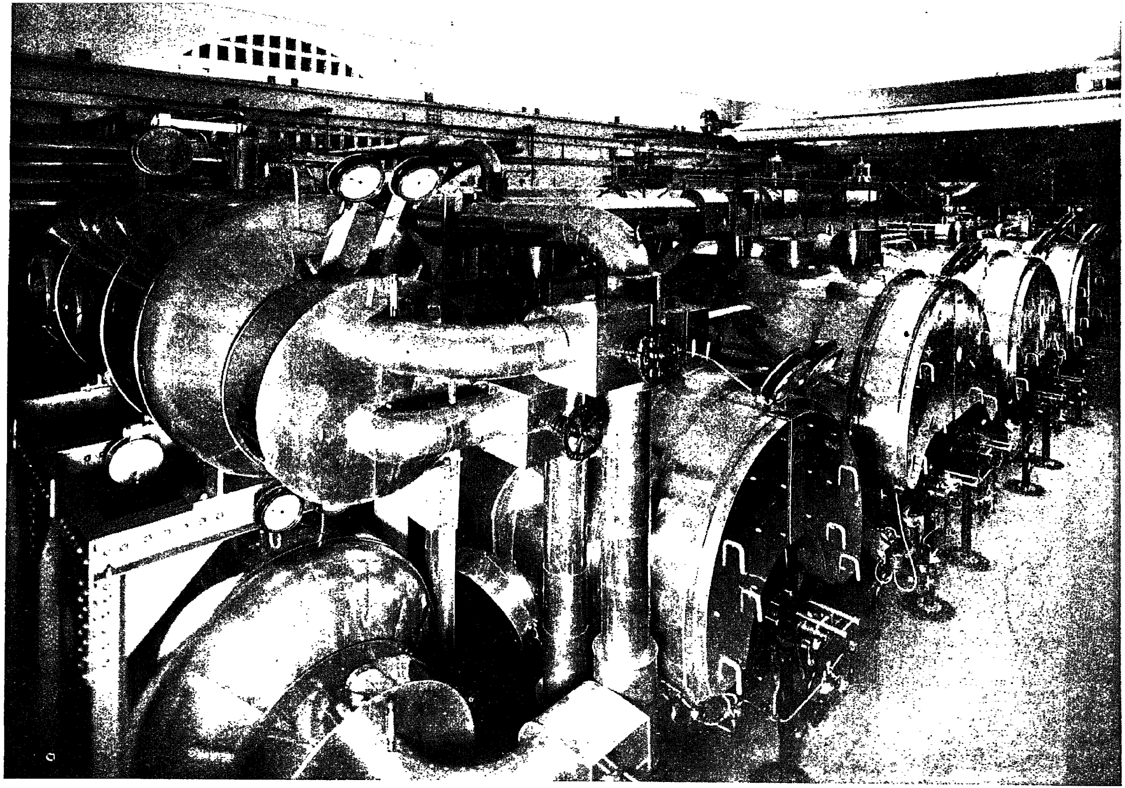
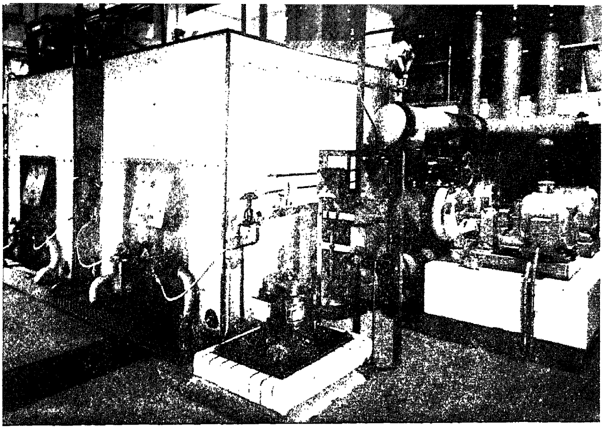
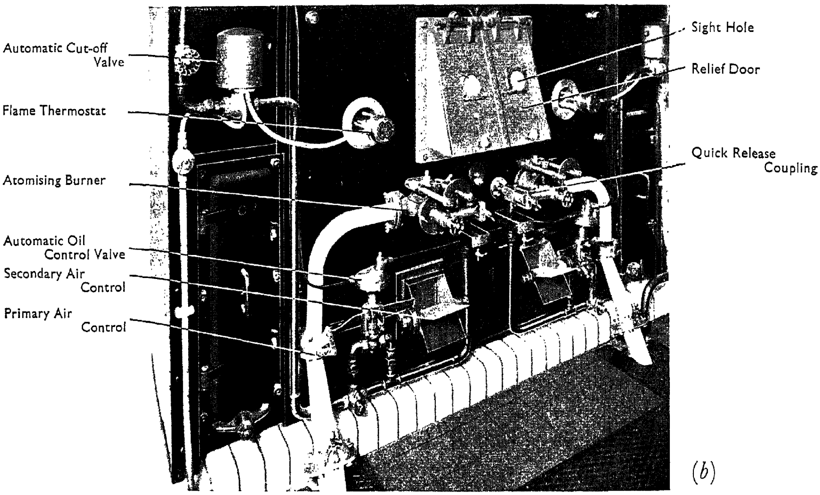


Plate IX. Oil-fired boilers at the Bank of England, now burning creosote-pitch (*see p. 164*)



(*a*)



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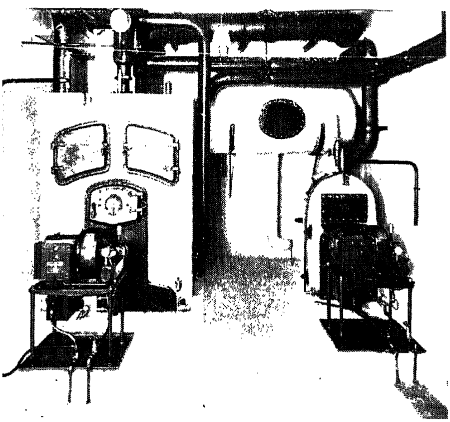


Plate X. Typical oil-fired heating installations.

(*a*) *India House, London. Thermostatic control of oil supply.*

(*b*) *South Africa House. Thermostatic control of oil and primary and secondary air.*

(*c*) *Completely automatic burners serving sectional heating boiler and steel domestic boiler.*

(*see pp. 164-5*)

CONTROLS

Control may be manual, semi-automatic, or completely automatic. Since oil is by its nature a labour-saving fuel, the ideal always aimed at is to make an oil installation completely automatic and dispense with attendance altogether. For this reason the first two types of control will be mentioned shortly and the completely automatic type discussed at greater length.

Manual Control comprises hand ignition and hand operation of the fuel valve and air dampers. For large plants with steady load this may be satisfactory, but may lead to wasteful firing for reasons already stated.

Semi-Automatic Control applies only to the latter two systems and denotes hand ignition, but automatic control of oil and air. This generally takes the form of thermostatic control operated from water temperature or steam pressure and is arranged to control the flame in two stages—full and minimum flame; the flame is never extinguished. Further controls with some systems regulate the primary and secondary air sympathetically with the oil flow, so maintaining a correct oil-air ratio and high CO_2 content over the whole range of operation.

A further essential control with the semi-automatic plant is one which shuts off the oil supply in the event of flame failure, so preventing unburnt oil being delivered into the combustion chamber. This operates with a 'fluestat' in the boiler flues or combustion chamber.

A still further control shuts off the oil in the event of failure of air pressure due to stoppage of the compressor or fan motor. Such controls are operated either electrically or by air or water pressure.

Completely Automatic Control—The completely automatic burner is one which is self-igniting and self-extinguishing, in addition to being thermostatically controlled. It takes the form of a self-contained unit which is now quite well known and has long passed the experimental stage. Plate Xc shows a small installation serving heating and domestic boilers in a London Bank.

The limit of output is more dependent on the boiler than on the burner, since, when the boiler is cold, the rapid expansion produced when a large burner is suddenly turned full on without any gradual warming up period may be so severe as to cause dangerous straining of the joints and plates. Sizes much over 1,750,000 B.T.U.'s per hour in one unit are thus not commonly recommended.

The number of makes and types of these units is too numerous to describe, but all, in the main, are of a pressure jet atomizing type, the combustion air being supplied at low pressure by means of a fan. The fan is generally arranged to deliver the full 100 per cent. of the air required. Practically all types employ a pump for injecting the oil through a jet under pressure, but the burner nozzle takes a variety of forms.

One motor is arranged to drive both the fan and the oil pump and is

mounted on a base plate with the various electrical control devices, safety switches, oil filter, oil heater, etc., around it.

Ignition is generally by means of an electric spark derived from a transformer also fixed to the unit, and with the spark gap electrodes carried through the air blast tube in front of the burner tip. The spark is generally arranged to cut off after the oil is alight so as to save wastage of the electrodes. An alternative method of ignition is by gas, a small pilot flame being kept constantly alight, and expanded for a period when the plant is switched on. Sometimes both gas and electric ignition are provided as a safeguard against failure.

The water temperature is controlled by a thermostat in the boiler flow, which acts by switching off the fan and consequently the oil supply; and in addition it is often convenient to provide a room control, the boiler-flow thermostat then acting as master; alternatively a 'Variostat' type of control may be used as referred to with solid fuel firing.

Such controls are sometimes arranged with a day and night setting. A clock control changes over on to the lower setting at a specified hour and returns to the higher one in the morning. This is sometimes further extended to remain on the lower level over the week-end when applied to offices, shops, etc.

The completely automatic system may be adjusted to a high state of efficiency, but it should be remembered that, when shut off, cold air may leak in through the fan and so cool the boiler down, with a corresponding wastage of fuel. Many plants employ means for preventing this loss and this safeguard is most desirable.

These plants are generally quiet in operation, and may be left unattended for long periods. They are frequently left running all night, cutting in and out in response to the thermostat.

An almost unlimited number of circumstances may be catered for with automatic control. For example, in a combined heating and hot-water supply system, hot water may be provided for baths, with radiators operating at a lower temperature in mild weather, this being effected by a thermostatically controlled valve in the water flow to the heating system, the oil burner control then being the master.

FUEL OIL STORAGE

It is usual where space permits to provide a storage based on about three to four weeks' running at full load. Except in special cases of small installations, a storage capacity of six tons should be considered the minimum, as this enables a 5-ton load to be received with a ton still in reserve. It is an advantage to have two tanks, so that one may be cleaned with the other in use. In large installations a greater number will, of course, be necessary to obtain adequate storage.

Table XXXII gives the consumption of various sized plants based on

four weeks' run at full load (twelve hours per day), from which the appropriate storage may be estimated.

TABLE XXXII

CAPACITY OF OIL STORAGE TANKS

Boiler Capacity B.T.U.'s/Hour	Approx. Oil Consumption per 4 Weeks; Full Duty for 12 Hours per Day at 66 Per Cent. Efficiency	Cu. Ft. Capacity based on Sp. Gr. ·9
250,000	3 tons	120
500,000	6 "	240
750,000	9 "	360
1,000,000	12½ "	500
1,250,000	16 "	640
1,500,000	19 "	780
2,000,000	25 "	1000
4,000,000	50 "	2000
6,000,000	75 "	3000
8,000,000	100 "	4000

Storage Vessels in Buildings—Oil storage vessels are of cast iron or steel, the former being always sectional, and the latter sectional, welded or riveted.

Sectional tanks are of necessity rectangular, and have their joints made with a special oil-resisting cement. Cast-iron sections are generally 2 ft. square, bolted together with the bolts inside. Those of steel are of either 1 metre or 4 ft. square plates, bolted externally.

Welded steel tanks are often used and may be either rectangular or cylindrical, brought into the building in one piece or welded *in situ*. The latter is necessary in confined situations.

Tanks deeper than 3 ft. should have iron access ladders inside. To allow proper inspection outside of the tanks they should have a walking way of 18 in. clear all round, and should be carried on steel joists bearing on sleeper walls about 12 in. high, the top of the joists being covered with sheet lead to prevent rusting.

Pipe work and tanks used in connection with oil should not be galvanized, as this is liable to be attacked. Welded pipe joints are much to be preferred to screwed.

Tank Rooms separated entirely from the boiler house by a brick or concrete wall are to be preferred, and are, in any event, called for where a building has to be licensed by the London County Council. Where the tank room has only a dwarf-wall separating it from the heating chamber the smell of oil pervades the whole room when manholes are opened and may enter the surrounding corridors and building.

An oil-storage room should have the lower portion oil-tight to hold the full capacity of the tanks, should they leak; this calls for an access door at a higher level with steps outside and inside. The L.C.C. may require an air lock to be provided to this door, also separate inlet and outlet ducts

direct from outside into the tank room. These ducts must have an area of about one square foot, be of metal, concrete or brick, one being carried near to floor level to serve as an inlet. Gratings to each duct are required at the upper ends. Access doors must naturally be fire-resisting.

Fig. 97 shows this arrangement, together with other details which will be referred to later.

Tank Fittings—Various fittings are required in connection with storage tanks, the principal ones being given below.

A manhole must be provided to each tank, a common size being 18-inch diameter. The joint with the tank top should be made gastight.

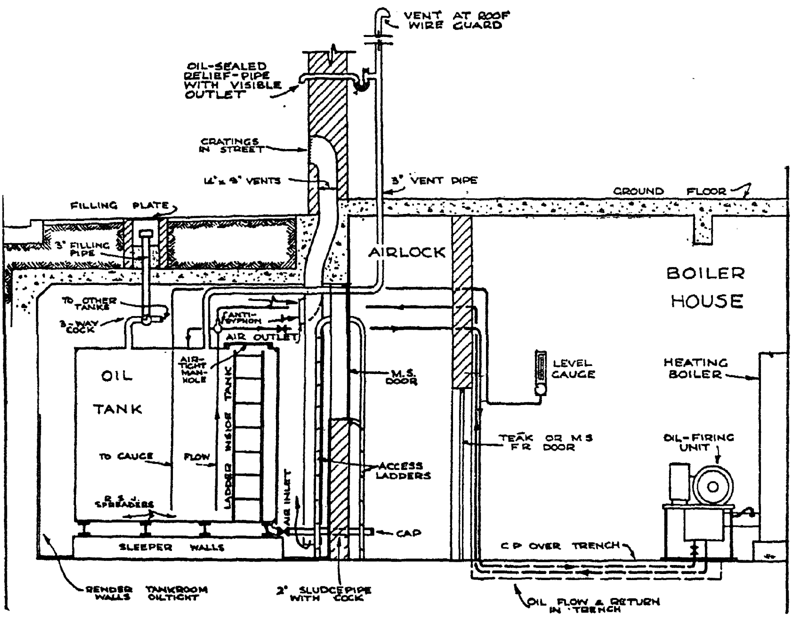


FIG. 97.—Typical Arrangement of Oil-Tanks, etc., for Heating Installation.

The filling terminal should be standard 3-inch male gas thread for hose coupling with gunmetal cap. From this a 3-inch. pipe connects to the tank with a steady fall. Where more than one tank exists a 3-way cock or set of valves will be required or one filling pipe per tank.

Vent pipe. A vent pipe from each tank is required, and should be 3 in. diam., carried separately up to the roof of the building and fitted with a wire guard.

It should be remembered that in the event of over-filling of a tank from a road wagon, delivering with compressed air or pump, the combined vent pipe arrangement allows oil to flow into the next tank through the pipe without obstruction. Where a separate vent to each tank is provided, this

is not possible, and the oil may rise in the vent pipe to a considerable height if the building is tall, so placing an undue pressure on the tank. Thus storage vessels should either be strong enough to withstand this head, which is probably uneconomical, or some means of relief should be provided. This may take the form of an oil seal branched from the vent pipe, as in Fig. 97.

It is unnecessary to point out that vent pipes should have a steady rise to the top, as any dip which might become filled with oil would obstruct the free passage of vapour.

The outlet connection should have a valve next to the tank and should be 3 or 4 inches above the tank bottom. The size is determined by the number and size of burners connected, each of which should again be separately valved.

If, under the L.C.C., the outlet has to take the form of a vertical pipe taken inside the tank and out of the top, an anti-syphon fitting is necessary at the highest point to prevent syphoning into the boiler house, such as might occur from a broken pipe joint below the level of the oil.

Another arrangement of the outlet connection is one in which a daily service tank is used. This should not exceed 100 gallons capacity. This tank is placed in the boiler house above the burners and feeds them by gravity. A pump, either hand or electric, is used to fill this from the main storage.

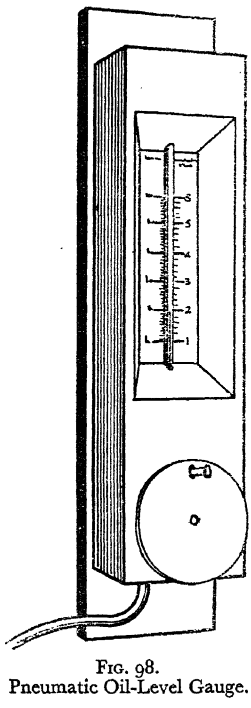


FIG. 98.
Pneumatic Oil-Level Gauge.

Sludge outlet. Water, being denser than oil, collects at the bottom of the tank, generally in the form of mud, being mixed with the sludge and solid matter which gradually settles out from the oil. After a few years of partial emptying and refilling of the tanks the accumulation obviously increases, and before there is any chance of its reaching the outlet it must be removed. A sludge outlet is therefore provided at the lowest point and is fitted with a valve. The size should be 2 in. and the end may terminate in the tank room, provided it is easily accessible, or in the boiler house, where it should have a screwed plug. The oil companies generally make arrangements for the collection and removal of sludge, as it cannot, of course, be put down the drains.

Incidentally, it should be pointed out that creosote oils, being more dense than water, cause the latter to float on the surface, and it is then important not to drain the tank to the level of the outlet, as oil and water in the burner are very troublesome.

Level Indicator. Gauge glass indicators, being fragile, are unsatisfactory. The best and simplest device is a pneumatic gauge, various makes of which are available, and one of which is illustrated in Fig. 98 (previous page).

This form of gauge allows the top of the tank to be kept gas-tight and conforms to all regulations.

Underground Storage Tanks—Where it is possible, tanks may be placed out of doors underground. This economizes building space and is a very safe and practical arrangement. If buried without an enclosing pit (as in Fig. 99), the tank should be properly protected with concrete or bituminous paint on the outside and the bottom of the excavation should be drained. A better arrangement is to construct a concrete pit so that inspection may be made all round. However placed, tanks or cylinders should have a slight fall to the sludge outlet for cleaning. Where the soil is waterlogged, it is generally necessary to anchor the tanks to a heavy block of concrete to prevent them 'floating' when empty.

The remarks made above regarding tanks in buildings apply in general to external buried tanks. The sludge removal from the latter is effected by pumping through a pipe carried nearly to the lowest point of the tank as in the figure.

FURNACE LININGS

The firebrick lining of oil-fired combustion chambers calls for a material which will withstand temperatures of 2500° to 3000° F. without fusing or premature disintegration. The brick, therefore, requires to be burnt to a temperature in excess of this, and several standard brands are available for the purpose.

One important point to watch in the building-in of this brickwork, particularly with a sectional boiler, is that the material is not carried solid up to the metal without an air-gap of half an inch or so being left around it. The great temperature in the combustion chamber causes considerable expansion of the bricks, and the air-gap allows freedom for this to take place without any strain being put on the boiler plates or joints.

SAFETY PRECAUTIONS

Fusible Links—One point not mentioned in the discussion on outlets from tanks is the provision of a fire valve. This generally takes the form of a lever type valve heavily weighted, kept open by a taut wire stretched across the boiler house and having a fusible link of low melting point alloy over each boiler front. Such an arrangement may be seen in Plate IV.

Should a fire occur, the rising flames or hot gases will melt the link and the valve will shut immediately, thus cutting off the supply of oil and minimizing further damage. A hand release is also provided for testing.

Further Safety Precautions—Insurance companies and licensing authorities often call for further devices for protection against oil fires, particularly on the large plants. These include the following:

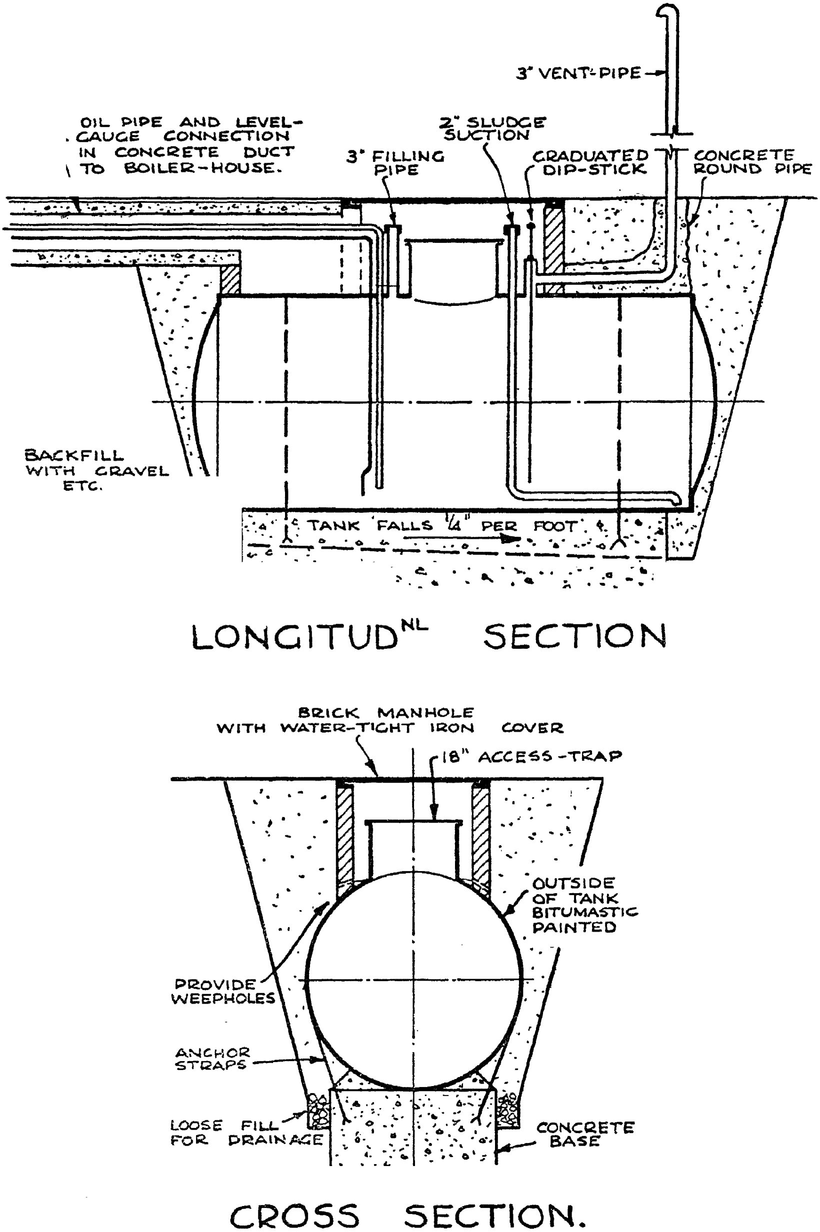


FIG. 99.—Arrangement of Underground Oil-Storage Tank.

Note. The vent-pipe, dip-stick, etc., are all shown on the centre-line for clarity, but could be arranged in the corners of the manhole round the circular access-trap.

(*a*) Boiler dampers to be removed or locked open with an indicating plate showing position of vane.

(*b*) Alternatively the burner valve and damper must be provided with an interlock operated by a common key which cannot be withdrawn from the damper lever until this is in the open position.

(*c*) A foam chemical extinguisher of portable or permanent type to be installed, serving both the boiler room and oil-tank chamber.

(*d*) In addition to the latter a curb around the boilers is called for to serve as a catchpit for escaping oil and the foam.

(*e*) Alternatively to (*c*) foam pipes to boiler house and tank chamber to which the fire brigade can connect their apparatus in the street.

(*f*) A sump in the oil-tank room provided with a ball float and electrical contact so as to give a warning signal immediately there is a leakage of oil from the tanks.

Creosote-Pitch Firing—Oil, being an imported fuel is, under war conditions, much restricted in supply, whereas the production of tar oils at home has greatly increased. The Government have, as a result, requested large consumers to change over to home-produced fuels. How permanent this policy will be remains to be seen, but, as more uses for crude oil are found, it may be necessary ultimately to prohibit its use for burning in boilers, in which case the tar oils will be the chief liquid fuel available.

Creosote-pitch is in abundant supply, and a great number of conversions have been made of existing oil-burning plants. This fuel is a solution of pitch in creosote; it is viscous, requiring continuous heating, and it burns with a clear flame very similar to oil, but with less soot and deposit.

A typical analysis of creosote-pitch is as follows:

Proximate	%	*Ultimate*	%
Fixed Carbon	15·71	Carbon	88·98
Volatiles	83·51	Hydrogen	6·64
Ash	0·18	Oxygen	2·39
Moisture	0·6	Nitrogen	0·59
		Sulphur	0·62
		Ash	0·18
		Moisture	0·6

Closed Flash Point	150°
Specific Gravity at 100° F.	1·163
Calorific Value, Gross	16,714 B.T.U./lb.
,, ,, Nett	16,131 ,,
Viscosity Redwood No. 1 at 80° F.	5000 secs.
,, ,, ,, at 100° F.	1000-1500 secs.
,, ,, ,, at 200° F.	100 secs.

The usual plant required for burning creosote-pitch in boilers includes:

(1) Storage tanks warmed to about 100° F. by electric heaters, or steam or hot water coils.

(2) Pressure pumps circulating the oil in heated pipes to the burners. The pipes are heated by outside electric soil heating cable or by a tracer pipe fixed adjacent, the whole being insulated together. Duplex filters are required at the pumps.
(3) Spring loaded valve on return of circulating system to maintain a pressure at the burners.
(4) Heaters near burners (electric or steam-heated) thermostatically controlled, to give a final temperature of 180° to 200° F. at the burners.

Oil-burner systems used with normal fuel oil are suitable for creosote-pitch, such as the low pressure air or pressure jet system. A special alloy for the jets used in the latter is available to withstand the erosive action of creosote-pitch. In the case of the former a good air pressure is required of 16-20 ins. w.g.

The brickwork inside the combustion chamber has to be more adequate than with straight oil, and preferably of high alumina type such as 'Nettle Brand'.

In operation it is imperative to remove the burners whenever shut down, and insert them at once in a bucket of Creosote A to prevent clogging.

Thermometers on tanks and at burners are necessary, together with pressure gauges on the piping and suitable flame failure equipment. Operating bellows must be of stainless steel, not copper.

The creosote-pitch is delivered in normal tank lorries not normally heated for short distances, but heated for long journeys.

Straight Pitch—The firing of straight pitch is confined to industrial plants. There are a number of systems developed by the various oil burner manufacturers, and it will be given no more than passing reference here. The pitch has to be melted in tanks and circulated at about 400° F. in pipes kept hot by h.p. steam or hot water tracer lines.

Straight pitch firing is not suitable for normal-sized heating use, owing to the complication and skilled attention necessary. Precautions are necessary to preserve the health of the operating staff.

CHAPTER VIII

Warming by Hot Water

In this country, with its relatively mild climate, hot water is usually the best compromise as a medium for heating, on account of its simple temperature control to meet variations in weather, and the absence of the parching effect commonly associated with steam and heated air. Warming by hot water therefore deserves detailed consideration.

The question of the choice and disposition of heating surfaces for any particular installation has previously been discussed, and it now remains to show how the amount of such surface necessary to meet a given heat loss may be established.

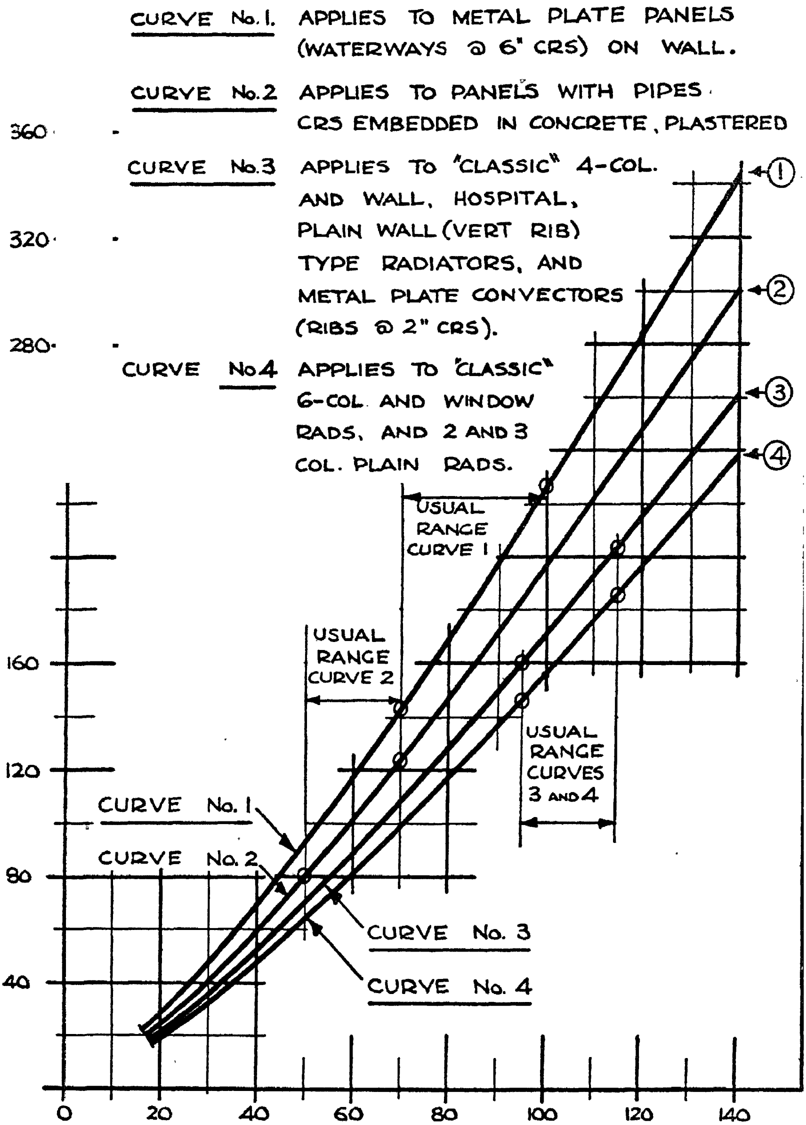
EMISSION FROM HEATED SURFACES OF VARIOUS FORMS

The total emission from a heated surface may be divided into radiant and convective components, of which the radiation is proportional to the difference of the fourth powers of the absolute temperatures of the radiating and absorbing surfaces. The convection varies considerably with the form and height of the surface, but if the heat emitted by convection at any one temperature is known, that at some other temperature will be proportional to the temperature difference to the power of 5/4 (approximately).

Due to the convection currents set up by any heated element in air, no surface gives 100 per cent. radiation, though a ceiling panel closely approaches it (about 90 per cent.). Panels on walls may have 60 per cent. radiation and 40 per cent. convection, and in floors 50 per cent. radiation and 50 per cent. convection. Radiators of the ordinary cast-iron type vary according to their convolutions, but transmit commonly 20 per cent. by radiation and 80 per cent. by convection.

Combined radiation and convection from most types of heated surface, when plotted for various water-air temperature differences, is found to follow a logarithmic curve. Knowing the transmission at any one temperature the total transmission (radiation plus convection) at any other normal temperature may be calculated approximately as proportional to the temperature differences to the power of 1·3.

Fig. 100 shows curves plotted on this basis for four main classes of heating surface. It will be noted that whilst those mostly radiant (curves 1 and 2) are superior in transmission to those mostly convected (curves 3 and 4) the differences are not considerable. The figure also shows the range of temperature over which the various forms of surface normally operate.



Temperature Difference 'θ' between Water and Air. In Degr. F.

Fig. 100.—Emission from Various Types of Heating Surface.

In using the curves the temperature of water to be taken in arriving at the difference between water and air, is the mean of the flow and return. This, however, is not necessarily the mean surface temperature except in the case of radiators where the drop through the metal is negligible. With metallic plate panels a water mean of 150° will give a surface mean of 140°, and with embedded panels a mean water temperature of 125° will give a surface mean of 90°-95°. Actually, of course, temperatures vary over the whole intermediate surface between the heating elements.

EMISSION FROM PIPES

Piping of cast iron or steel, when used as a heating surface, is commonly placed near the floor round the walls, or overhead under an exposed roof or skylight. Frequently two, three or four pipes placed vertically above one another are required to provide the necessary heating surface, though where more than two pipes are used, the masking effect of the adjacent surfaces becomes important and the emission is reduced.

It is sometimes convenient, and architecturally permissible, to provide the main heating surface in the form of radiators with the main pipes serving them left exposed. In this case the heat emission from the pipes is taken as assisting in the general heating, and greater economy is achieved than if they are buried in a trench.

Table XXXIII gives the emission in B.T.U.'s per foot run of the commercial sizes of pipe for various temperature differences. The figures given are for single pipes. For pipe coils with spacing not less than one pipe diameter apart, deduct from the figures 5 per cent. up to 2 ft. high, and 10 per cent. up to 4 ft. high.

The following example will illustrate the use of this table:

Water flow temperature—180° F. } mean „ return „ —140° F. }	160° F.
Air in room	60° F.
Difference	100° F.
Heat loss of room	22,500 B.T.U.'s per hr.
Length of pipe possible around room	100 ft.

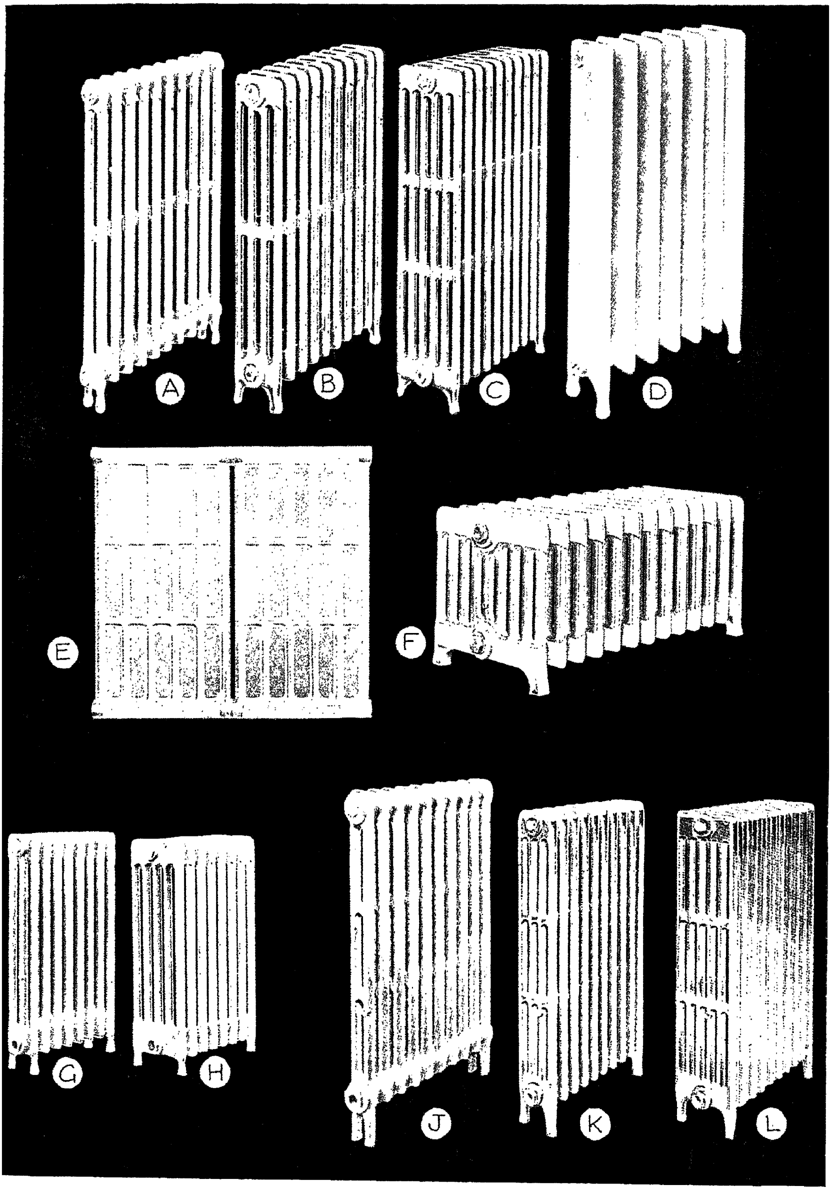
$$\text{B.T.U.'s per foot} = \frac{22{,}500}{100} = 225.$$

With 100° difference (from table)	a 4-in. pipe gives 234 B.T.U.'s per ft. or two $1\frac{1}{2}$-in. give 220.

The table may also be used for estimating the losses from mains.

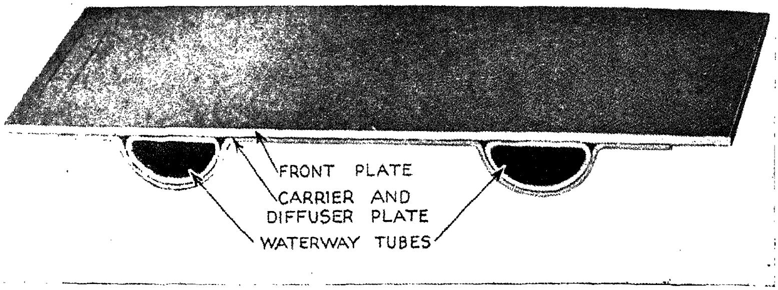
RADIATORS

Emission from Radiators—Curves 3 and 4, Fig. 100, apply to the types of radiators in common use. Several of these types are illustrated in Plate XI (facing page).

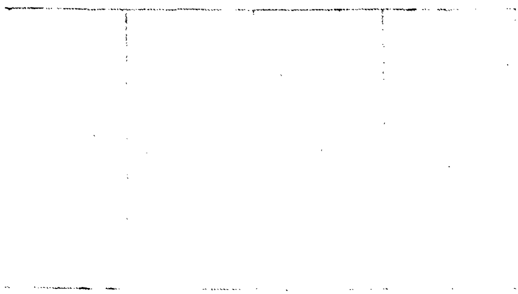


XI. Types of radiator (*see p. 176*)

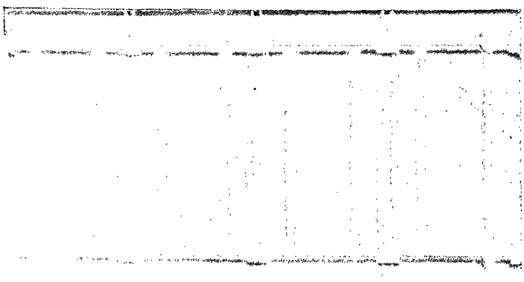
(*A*) *Ideal 'Classic'—2 column*
(*B*) *Ideal 'Classic'—4 column*
(*C*) *Ideal 'Classic'—6 column*
(*D*) *Ideal 'Hospital'*
(*E*) *Ideal wall-type*
(*F*) *Ideal window-type*
(*G*) *Beeston 'Royal'—3 column*
(*H*) *Beeston 'Royal'—5 column*
(*J*) *Crane 'Pall Mall'—2 column*
(*K*) *Crane 'Pall Mall'—4 column*
(*L*) *Crane 'Pall Mall'—6 column*



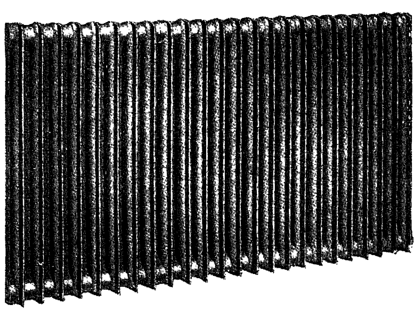
(*a*)



(*b*)



(*c*)



(*d*)

Plate XII. Types of radiant panel (*see p.* 181)

(*a*) *Cross-section of 'Solray' border panel* (*steel*)

(*b*) *Ideal 'Rayrad' No.* 15 (*front view*) (*cast iron*)

(*c*) *Ideal 'Rayrad' No.* 15 (*back view*) (*showing waterways*)

(*d*) *Ideal 'Rayrad' No.* 24 (*ventilating*) (*back view*)

TABLE XXXIII

EMISSION OF IRON PIPES IN B.T.U.'S PER LINEAL FOOT PER HOUR.*

Nominal Size of Pipe Inches (Int. Dia.)	Sq. ft. Heating Surface per Lineal ft.	B.T.U./hr./sq.ft./1°F. at 100° Difference	Temperature Difference (° F.) between Air and Mean Water						
			60°	70°	80°	90°	100°	110°	120°
½	0·21	2·67	29	35	42	48	56	63	71
¾	0·28	2·50	35	44	53	61	70	79	88
1	0·33	2·40	41	50	60	70	80	90	101
1¼	0·43	2·30	49	61	73	85	98	110	123
1½	0·50	2·20	56	69	82	96	110	124	139
2	0·62	2·11	70	85	101	118	135	153	171
2½	0·75	2·05	79	96	115	135	154	174	194
3	0·92	2·01	95	116	138	161	184	208	234
3½	1·05	1·98	107	130	156	182	208	234	263
4	1·19	1·96	120	147	175	204	234	264	297
5	1·46	1·93	144	176	212	247	282	318	354
6	1·75	1·89	170	208	249	291	331	373	415
7	2·00	1·87	193	235	280	326	374	422	475
8	2·26	1·86	216	264	315	367	420	474	534
9	2·52	1·85	240	294	350	407	467	526	593
10	2·82	1·84	268	327	390	454	520	596	660
12	3·35	1·84	322	394	470	546	626	707	795

* Table is for bare pipes. For lagging 75% efficient, divide above figures by 4.

TABLE XXXIVA

RADIATOR EMISSIONS IN B.T.U.'S PER SQ. FOOT PER HOUR (AS GIVEN BY MAKERS).
RADIATORS PLACED 1½ IN. OR MORE CLEAR OF WALL

(*a*) Ideal.

Type	Temperature Difference in ° F. between Air and Mean Water					
	70	80	90	100	110	120
Neo-classic No. 2	116	139	162	185	208	234
,, No. 4	106	128	149	170	192	215
,, No. 6	100	120	140	160	180	202
Hospital 3 in.	116	139	162	185	208	234
,, 5¾ in.	99	119	138	158	178	199
,, 7¼ in.	94	113	131	150	169	189
Neo-classic window	99	119	138	158	178	199
Classic wall	106	128	149	170	192	215
Plain wall (horizontal)	100	120	140	160	180	202
,, ,, (vertical)	64	77	90	103	116	130
Plain Single-column	100	120	140	160	180	202
,, Two-column	98	117	136	156	176	197

(*b*) Crane 'Pall Mall' types as for Ideal 'Neo-classic'. (Nos. 2, 4, 6, Hospital, window and wall.)

(*c*) Beeston 'New Royal' types as for Ideal 'Neo-classic' No. 2 and No. 4. For 3- and 5-column types, interpolate between Nos. 2, 4 and 6. Hospital, wall and window types as Ideal.

Table XXXIVA (p. 177) gives the coefficients for various temperature differences in greater detail. A mean water temperature of 160° is usually allowed, which, with a 40° drop through the system, normal with gravity circulation, gives a flow of 180° and return of 140°. With 30° drop, and pump circulation, these become 180° and 150°, giving a mean of 165°. In special cases these temperatures may be raised, with resulting economy. For preliminary calculation it will be seen from the table that a figure of about 160 B.T.U.'s per sq. foot is an average for general use.

Radiators are generally best placed under windows, both for architectural and technical reasons. Architectural because here they do not mar an otherwise unblemished wall surface, and technical, since this is the point of maximum heat loss. They counteract down draught, and do not cause dirty marks on the walls, since the window only is above.

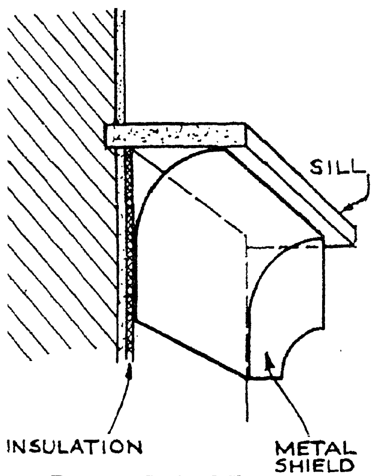


FIG. 101.—Sectional Elevation of Metal Radiator Shield.

Shelves over Radiators—It is frequently desirable to carry the sill of the window over the top of the radiator, the latter being placed in a recess in the wall so as to save floor space.

Similarly, when placed against a wall, not under a window, the provision of a shelf over a radiator is necessary to prevent, as far as posssible, the smoky marks above it, which would certainly otherwise occur. The jointing to the wall should be sound, as the slightest crack will allow a black mark to form, and end shields are necessary to prevent markings at the sides. A useful shield is one made of one piece of metal with the angle curved, and to the top of this, timber, marble, or other material may be screwed as in Fig. 101.

Provided such shelves are not less than 3 in. above the radiator the reduction in transmission caused by them is so slight that it may be ignored. In fact it may be argued that, as the uprising current is baffled to some extent by a shelf, the layering of hot air near the ceiling is reduced, with consequently better diffusion at lower levels.

Radiator Grilles—Where the appearance of radiators cannot be tolerated on architectural grounds, grilles of painted iron, bronze, hardwood or other material are commonly used to enclose them. These not only reduce the heat transmission, thereby calling for increased surface, but are in themselves costly and dirty. Means should be provided for their removal for cleaning, but even so it is generally found that the interior spaces are seldom brushed out.

It is desirable, therefore, to avoid these enclosures, if possible, by adopting other forms of heating surface, such as one of the panel types, and it is often questionable whether these are any more costly when the expense of the grilles is taken into account.

Cases exist, however, where radiators in grilles present the only solution, and it is then a question of how much extra surface is required to maintain the same temperature. This largely depends on the form of grille, and additions from 20 per cent. to 33 per cent., depending on the amount of free opening left, are necessary. It should be remembered that with a suitable design of grille it is possible to produce a 'flue' effect and so increase the transmission from a radiator, as in the case of the convector (see below).

Painting of Radiators—It has already been mentioned that the best radiating and absorbing surface is dead black, and that the painting of a surface white (such as a roof) will reduce the absorption and radiation considerably. Whilst this is true of high temperature radiations, such as is received from the sun, and is important when considering the material for a roof, it does not apply at low temperatures. Dr. Margaret Fishenden* states that the result of researches carried out by various authorities appears to confirm that radiation from various painted surfaces, glossy or matt, including black, green, brown, red, ivorine, and white, is indistinguishable at temperatures below 212° F. This is fortunate, since it means that any colour may be used with impunity.

Polished metallic surfaces are, however, bad *radiators* of heat (even at these low temperatures) (in some cases only 10 per cent. as efficient as the unpolished surface), as are the metallic paints such as aluminium, bronze or gold, though the *convection* is not of course affected.

The use of such metallic paints reduces the radiation component by about 50 per cent. Assuming that an ordinary radiator emits 20 per cent. by radiation, a 10 per cent. total reduction may be expected. It is stated that a coat of varnish over a metallic paint brings the emissivity back almost to normal.

This is not the place to discuss the reasons why polished surfaces do not radiate and absorb heat so readily as painted ones, or why certain surfaces are better emitters at one wavelength than at others, even if the knowledge were available. These matters are bound up with the whole theory of radiation, which is at present undergoing rapid development, and the observed facts are all that concern us here.

Convectors—Two types of convector are shown in Plate XIV (p. 183).

Type (*a*) is of finned copper, and is designed for fixing in a wall recess. Free-standing cabinet types are also available. The emission as stated by makers (Messrs. British Trane) with water at 160° F. mean (air at normal temperatures) is from 150 B.T.U.'s per sq. ft. on gravity systems to 185 B.T.U.'s per sq. ft. on pumped systems with water at 180° F. The heating

* *Calculation of Heat Transmission*, p. 19.

surface is given as Equivalent Feet Super, and varies with the height for any one size of unit.

TABLE XXXIVB

CONVECTORS (BRITISH TRANE)

Length	Width $5\frac{9}{16}''$					Width $10\frac{13}{16}''$				
	Height					Height				
	42″	36″	30″	24″		42″	36″	30″		18″
	Equivalent sq. ft. Htg. Surface					Equivalent sq. ft. Htg. Surface				
12	10·3	9·9	9·3	8·3	6·7	17·7	17·2	16·0	14·3	10·5
18	17·5	16·8	15·9	14·3	11·5	30·0	29·2	27·4	24·5	18·0
24	24·7	23·7	22·5	20·3	16·3	42·3	41·2	38·8	34·7	25·5
30	31·9	30·6	29·1	26·3	21·1	54·6	53·2	50·2	44·9	33·0
36	39·1	37·5	35·7	32·3	25·9	66·9	65·2	61·6	55·1	40·5
42	46·3	44·4	42·3	38·3	30·7	79·2	77·2	73·0	65·3	48·0
48	53·5	51·3	48·9	44·3	35·5	91·5	89·2	84·4	75·5	55·5

Other widths are $3\frac{9}{16}''$ and $7\frac{9}{16}''$. For fuller details see makers' lists.

Type (*b*) is of cast iron in a sheet steel casing. The unit may be fixed at various angles to give a variety of widths. The emission is given by makers with water at 170° mean, air at 65°, pumped circulation as follows:

TABLE XXXIVC

CONVECTORS (CRANE)

Length ins.	Width $4\frac{1}{4}''$			Width $7\frac{7}{8}''$			
	Height			Height			
	36″	24″	18″	36″	24″	18″	
12	1215	1075	955	1660	1460	1305	B.T.U.'s per hour
24	2440	2155	1890	3310	2930	2625	
36	3665	3235	2820	4960	4400	3945	
48	4890	4135	3760	6610	5870	5265	

For fuller details see makers' data.

Indirect Radiators. These are convectors composed of gilled pipes for placing below floors or in hidden recesses with grilles. They are largely superseded by the convectors described above and by other forms of heating. Details of emissions depend on the position and height of 'flue', and are given in standard makers' lists.

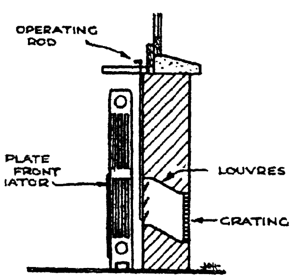


FIG. 102.—Radiator Fresh Air Inlet.

Fresh-Air Inlets to Radiators—Fresh-air inlets are frequently provided behind radiators as in Fig. 102. Their effect is to increase the heat transmission by some 20 per cent. to 30 per cent., which is enough to counterbalance the reduction in efficiency caused by enclosing grilles. When such inlets are used a baffle plate is necessary in front of the radiator to prevent the

direct in-flow of cold air, and means for closing the grating by adjustable louvres or a hit-and-miss register should be provided.

These inlets, admitting air straight from outside, tend to make the radiators and their enclosures dirty. It is, moreover, frequently found that they are stopped up on account of the draughts which otherwise occur with strong winds on that particular face. For these and other reasons they have fallen somewhat into disfavour, though their principle is sound.

Another type of baffle-plate provided with the 'Crane' radiators consists of small shields slipped in between the sections, giving a very neat effect.

RADIANT PANELS (METAL)

[See Plate XII (facing p. 177).]—Owing to the relatively high temperature at which these may be operated, the emission per square foot from curve 1, Fig. 100, will be seen to be from 169 to 225 B.T.U.'s per hour per sq. ft. of exposed surface.

With a standard cast-iron Rayrad section 30 in. high by 16 in. wide, giving $3\frac{1}{3}$ sq. ft. frontal area and with a coefficient of 225 (at 100° diff.) the transmission per section will be 750 B.T.U.'s.

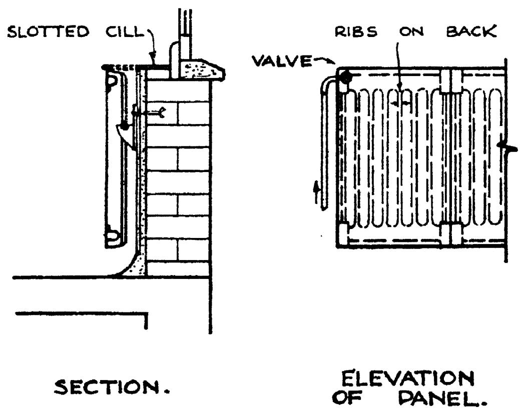


FIG. 103.—No. 24 Rayrad.

Table XXXV gives makers' transmissions for various types of Rayrad. It should be explained that Nos. 15, 16, 21 and 23 are variations of the plain flat type, completely of cast iron. No. 35 is a cast-iron skeleton with a steel front plate which may be cut to any reasonable size. No. 24 is a ventilating type shown in Fig. 103, having closer waterways and ribs at the back at 2 in. centres, thus providing an increased amount of convection. The chief advantage of this type is its compact size for a given rating.

In the ceilings of rooms less than 10 ft. high it is undesirable for metallic plate panels of any type to be used at temperatures of more than 120° on the surface, corresponding to a water mean of 130°, as above this the intensity of heat radiated on to the head is unpleasant.

TABLE XXXV

ACTUAL EMISSION OF IDEAL 'RAYRADS' IN B.T.U.'S PER SECTION PER HOUR FOR 100° TEMP. DIFFERENCE*

Rayrad	No. 35					Nos. 15, 16, 21, 23			No. 24		
Width of Section - -	12″					16″			16″		
Height of section, inches „ plate, inches -	30 36	24 30	18 24	12 18	12 13	30	24	18	30	24	18
Wall - - - - -	645	535	425	315	255	750	610	495	2060	1630	1200
Ceiling - - - -	480	400	320	240	190	570	467	373	—	—	—
Floor - - - - -	680	565	450	335	265	785	640	520	—	—	—

* For other temperature-differences add or deduct for each 10° F. variation:

No. 35	- - - - -	$\frac{1}{9}$ of rating given.
Nos. 15, 16, 21, 23	- - -	$\frac{1}{8}$ „ „
No. 24	- - - - -	$\frac{1}{7}$ „ „

Metal plate panels in the ceiling must be jointed to the plaster at the edges, and to prevent a crack appearing a cover strip is usual. This is not always architecturally desirable, and for this reason the embedded panel is to be preferred. Metal plates are much more adaptable to wall positions, though it will generally be found that their length considerably overruns the spaces under the windows, unless of the ventilating type, No. 24.

Steel plate panels (see Plate XII) are adaptable to almost any shape, size and position, but are best concentrated under or over the windows, where the major heat loss occurs. This type has also been developed for warming skirtings, floor-borders and cornices. The transmission rates may be taken from curve No. 1 in Fig. 100.

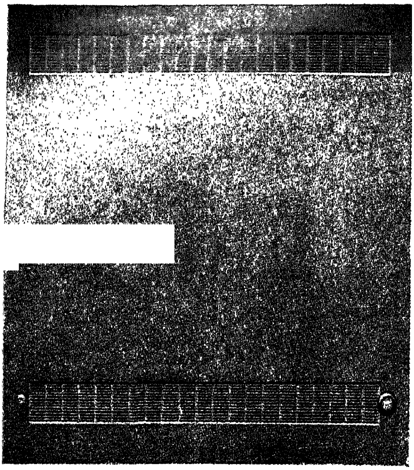
It is important that where radiant panels of any type back on to exposed walls, the rear surface should be insulated with fibre-board or similar material to reduce direct transmission losses. The same applies to all types of metal ceiling panels.

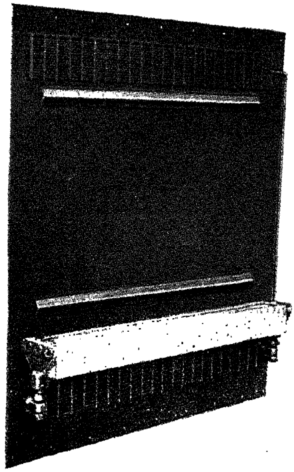
EMBEDDED PANELS

The system has been described in Chapter IV. Plate III shows panel heating during construction, and Plate XIII the areas heated when the system is applied to a Banking Hall.

The greater the spacing apart of the water tubes in an embedded panel, the greater, up to a point, the transmission will be per foot run of pipe, but the smaller the transmission per sq. ft. of surface, since a greater surface is available for radiation for a unit of length. Results of tests carried out with pipes embedded in a concrete ceiling at 4½ in., 6 in., 9 in., 12 in., and 18 in. centres show the transmissions per foot run of pipe and per sq. ft. of surface given in Table XXXVIA.

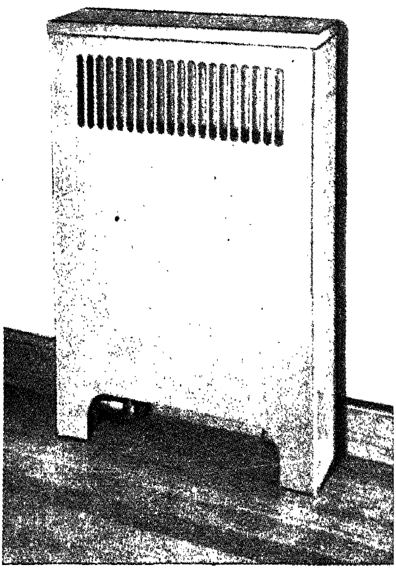
Plate XIII. Concealed panel heating applied to extensive areas of floor and ceiling at the Head Office of Glyn Mills Bank, London (see p. 182)

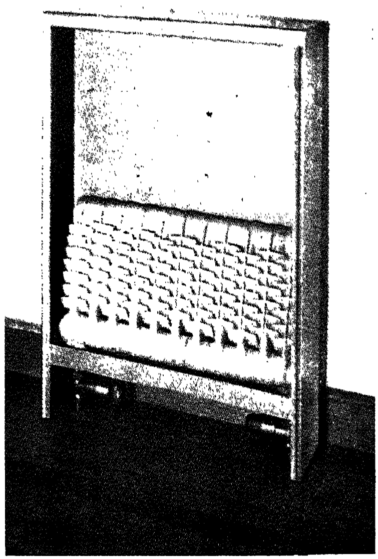




Front view *Back view*

Type (a). Trane Convector for wall recess





Front view *Interior view*

Type (b). Crane Convector

Plate XIV. Convector types (see p. 179)

TABLE XXXVIA

HEAT-EMISSION FROM EMBEDDED CEILING PANELS WITH VARYING PIPE SPACINGS

60° Diff. Mean Water-Air

Centres of Pipes	½-in. Copper Pipe		¾-in. Copper or ½-in. Iron Pipe	
	Per Ft. Run	Per Sq. Ft. of Surface	Per Ft. Run	Per Sq. Ft. of Surface
4½ in.	38	102	43	115
6 in.	45	90	50	100
9 in.	47	63	52	70
12 in.	49	49	54	54
18 in.	51	34	57	38

(See Table XXXVIB for recommended emissions.)

A spacing of 6 in. appears to be a good compromise between excessively large areas of heated surface and a reasonably high transmission rate.

A temperature drop of 15° F. is usual with embedded panels. This is less than commonly taken for radiator systems on account of the lower temperature of the flow and the necessity for providing a fairly uniform distribution of heat over the whole surface.

The maximum flow temperature taken for this system is 135° F., giving a return of 120° and a mean of 127½°. Normally a lower temperature suffices, about 10°-20° below this.

Floor Panels—When in floor screed, insulating coverings of wood, cork or rubber are unsuitable. Hard materials are liable to crack unless divided into small areas. This is naturally always done with marble, stone and tiles, but can also be arranged with terrazzo and other hard finishes and is to be recommended. A carpet placed over heated floor panels does not appear to reduce the transmission, as presumably the small fibres all become heated by conduction.

It is found to be unpleasant and tiring to sit for long with the feet in contact with a warm floor, even though the surface may be at no more than 80°, and for this reason floor heating surfaces are usually run at a lower temperature by wider spacing of the pipes.

The maximum floor-temperature to give comfort with continuous use is about 70°. Such a surface-temperature will be obtained with a mean water-temperature of about 115°, and pipes at 15 in. centres, giving a transmission of 20 B.T.U.'s per sq. foot for a room temperature of 60°. A safe rule for floor heating is to allow 2 B.T.U.'s per 1° difference between floor-surface and air temperatures.

Where floor heating is adopted, it is generally found necessary to cover the whole area of the floor with embedded pipes, varying the spacing according to the amount of heat required. Such systems have been successfully used in schools, particularly those of the 'open-air' type.

Floor panel coils are best made of copper, to avoid rusting due to water from washing of floors, etc., entering through cracks.

Floor panels with normal spacing of pipes and higher surface temperatures may be used as borders to rooms, and in entrance halls, etc., where persons do not sit for long periods with their feet in contact with the floor.

Wall Panels—Pipes may be embedded in walls and covered with plaster, as in the case of ceilings, or with marble or other finish not liable to cracking. In this position, the transmission is increased by about 40 per cent. over the ceiling coefficient, due to the increased convection from the surface.

Ceiling Panels—The normal (and often the most convenient) position for embedded pipe panels is in the ceiling. Here they are embedded in the concrete soffit of the floor slabs during construction and are subsequently finished with a special plaster made with a finely ground hydrated lime, sharp sand and long cowhair. The setting coat mesh has a scrim of $\frac{1}{8}$ in.

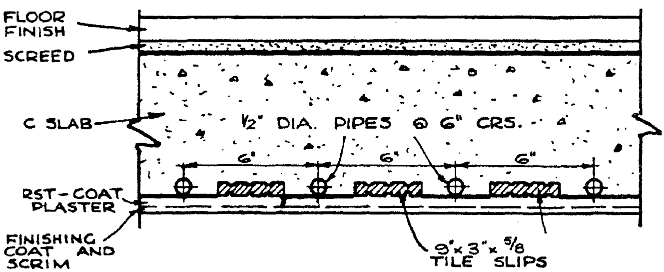


FIG. 104A.—Detail of Embedded Ceiling Panel in Reinforced Concrete Slab.

coarse hessian trowelled into it to prevent surface cracking. A specification of plastering most suited to panel heating has been prepared and should be carefully followed for successful results. It is absolutely essential that the first coat should be allowed to dry for three days or more before the second coat is applied, and for a further fourteen days to elapse before heat is applied.

Another point of great importance is the key provided on the concrete for the adhesion of the plaster. This requires to be as good as possible and is very suitably provided by laying grooved tile slips between the pipes, or by placing grooved rubber sheeting over the shuttering before the pipes are laid, this being subsequently removed, so leaving a deeply serrated surface (see Plate III, facing p. 49). Hacking is objectionable as giving an insufficient key and involving risk of damage to the panel pipes.

A section of an embedded ceiling panel in a reinforced concrete slab is shown in section in Fig. 104A.

Where a heating panel is placed in the soffit of a concrete slab, obviously the whole mass is warmed as concrete is a good conductor. A proportion of the heat will thus be transmitted upwards.

The amount of upward emission may be taken from Table XXXVIB, which also gives the downward and other emissions recommended for use.

TABLE XXXVIb

EMISSIONS RECOMMENDED FOR PANEL SURFACES. PIPES ½-IN. BORE

Mean Water 115° to 120° F.

	Emissions in B.T.U.'s per sq. ft. per hour Air Temp. in Room, ° F.			
	55	60	65	70
Ceiling panels (pipes at 6-in. centres)				
Downward, all cases - - - - - -	75	70	65	60
Upward with concrete slab, solid finish over	23	21	20	18
Upward with concrete slab boards and battens over - - - - - - -	15	14	13	12
Upward in roof slab, insulated over, say, 2-in. cork, 30° outside - - - - -	8	7	7	6
Wall panels (pipes at 6-in. centres) - - - -	110	100	90	80
Floor panels (pipes at 15-in. centres) - -	30	20	10	—
„ „ „ 6-in. „ - -	125	110	100	90

Where serving the ground floor of a multi-storey building this upward transmission should be allowed for by estimating the panel surface for that floor on the basis of the downward emission only, or under-heating of this floor may occur. With intermediate floors the upward heat balances the reduced heat downward.

Considering this aspect for a moment, it will be seen that if 30 per cent. of the heat is transmitted upward, the floor surface above is liable to become unpleasantly warm. Thus with an upward transmission from the panel of, say, 20 B.T.U.'s per sq. foot on the basis of 2 B.T.U.'s per 1° difference stated above, there will be a temperature increase on the floor above of approximately 10°, which, if the air is at 65°, will give a surface temperature of 75°. This is as high as desirable for comfort, and this is one of the reasons why it is usual to limit the water temperature in ceiling panel systems in multi-storey buildings to about 115° mean.

A number of different applications of embedded panels are possible for floor slabs and false ceilings of various types, but these are too numerous to detail here. It may be said, in general, that, provided adequate care and experience are brought to bear on any particular case, there is no type of ceiling construction to which embedded heating panels cannot be satisfactorily applied.

The normal position for ceiling panels is in the form of a broad band within one or two feet of the outside wall and windows. Thus any occupant sitting nearest to the greatest exposure is receiving the maximum of radiation.

A question often asked is, 'What about expansion of the pipes, and won't the ceilings crack?' The answer is that when embedded in a solid slab the pipes are restrained from expanding. This restraint brings into play compressive stress and strain in the pipe and tensile stress and strain in the concrete. When low temperatures are employed, as recommended, it can be shown that the stresses induced are too low to cause cracking of

concrete or damage to pipes. This is, in fact, borne out in practice, as hundreds of examples can be cited where this type of heating has been applied without cracking.

A test was carried out some years ago at the Authors' instigation with a sample panel of pipes embedded in concrete, and plastered in the normal way. Boiling water and iced water were alternately passed through the coils in an endeavour to produce cracking, but without success. Whilst this is an interesting experiment, it is not suggested as applicable in practice with slabs of great expanse. It does, however, indicate a considerable factor of safety at the small ranges of temperature normally employed.

In the early days of panel heating, the concrete floors were cast first, and panels of compo pipe were then slung up and fixed thereto and subsequently plastered. Here the normal insecurity of plaster applied to a concrete ceiling was much enhanced by the expansion and contraction of panels plus plaster, and separation was bound to occur. This type of pipe also had a low yield point and suffered by being strained beyond this limit with many reversals of stress.

None of these objections apply when copper or steel pipes are embedded in the concrete slabs above the plane separating them from the plaster.

Ceiling Panels in High Rooms—The effectiveness of ceiling panels is reduced with increasing height of rooms. An arbitrary allowance for this is to add 1 per cent. to the heating surface for every foot of height above 15 ft. In addition it is desirable to bring the panels for unusually high rooms nearer to the centre of the ceiling so as to reduce the amount of heat falling on the exposed walls and windows. High rooms commonly have tall windows, and it is desirable to check the down-draught from these by a band of panels or convection heaters under the sills.

'PANELITE'

A proprietary version of panel heating under the above trade name consists of pipe coils enclosed in a split tubing of asbestos cement as shown in Fig. 104B. This arrangement allows freedom for expansion and contrac-

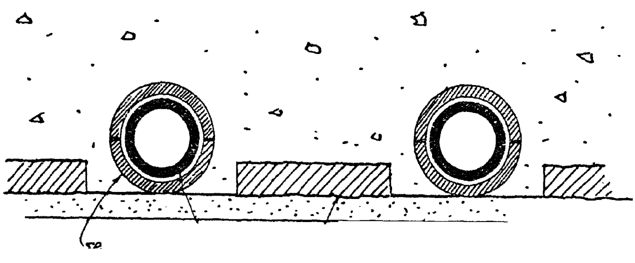


Fig. 104B.—'Panelite' Heating as applied to Concrete Slab.

tion, and consequently high water temperatures may be used then in the solid embedded type. The coils can be installed in concrete or hollow tile slabs, or laid below floors or recessed into walls.

COMFORT EFFECT WITH RADIANT SURFACES

The radiation of large flat surfaces, covered by curves 1 and 2, Fig. 100, is often given an increased value on account of the so-called 'comfort effect'. When, as is usual, however, a guaranteed temperature (as determined by an ordinary thermometer) has to be provided, this effect may not be allowed for and the actual transmission in B.T.U.'s per sq. ft. taken from the curves or from the tables is the only safe basis.

It is clear that such an effect exists, but with low-temperature surfaces under stable conditions it is slight. As the surface temperature becomes higher, this factor grows increasingly important. Further research is necessary to enable the comfort effect of radiant surfaces to be calculated on a basis such as the Equivalent Temperature Scale already referred to.*

* See also paper by H. Bruce, *I.H.V.E. Journal*, 1936–7, p. 427; and the *Heating and Ventilating Engineer*, August 1942, p. 52.

CHAPTER IX

Warming by Hot Water: Piping, and General Considerations

Having selected the type, position, and size of heating surface, and marked this on the plans of the building in question, the next stage is to determine how this shall be connected to the boiler with pipes in the most economical and effective manner, and how the water shall be circulated, i.e. whether by gravity or by a pump.

Gravity Circulation is suitable for the heating of houses and small buildings generally, or even for larger buildings where the ratio of height to horizontal run is high. It is unsuitable for extensive buildings where considerable horizontal runs are necessary with little or no radiation above the boiler level.

Pump or Accelerated Circulation is suitable for all types of building except possibly residences where the running of a pump might be considered inadvisable or unnecessary. It has already been pointed out that extensive ranges of buildings may be served from one set of heaters with pump circulation, as also may buildings having all the radiating surface at or below the level of the boiler. (This sometimes occurs in the case of theatres where basement space is valuable and the boiler house is therefore placed on the roof.)

The advantages of pump circulation may be summarized as follows:

(*a*) Smaller pipes, valves, and radiator connections may be used, giving reduced cost, neat appearance, and low heat loss from mains.

(*b*) Rapid heating up from cold due to high velocity of water through pipes.

(*c*) Quicker cooling down since less heat is stored in the mains.

(*d*) As a result of the above a saving in fuel consumption should be made as compared with gravity circulation, especially in buildings intermittently heated. As against this the cost of electricity for operating the pump has to be allowed for.

The exact line of demarcation between systems in which gravity circulation is sufficient and those in which a pump is desirable, is difficult to define, but the tendency now is towards the increased use of artificial circulation, even on the smaller installations.

Self-Accelerated Systems—Mention should perhaps here be made of accelerated systems of the past in which a pump was not employed. Most of these depended for their operation on steam generated by the heating boiler, or independently, and have fallen into disuse, partly because of their complication and uncertain results, and partly because of the sim-

plicity and cheapness of the centrifugal pump and ease of obtaining electric power supply even in country districts.

SYSTEMS OF PIPING

Systems of piping may be classified as follows:

(*a*) One-pipe ring main (Fig. 105).
(*b*) One-pipe drop (Fig. 106).
(*c*) Two-pipe rising (Figs. 107 and 108).
(*d*) Two-pipe drop (Fig. 109).
(*e*) Ladder (Fig. 110 (*a*)).
(*f*) Irregular (Fig. 110 (*b*)).

Comments on the use and application of each system appear adjacent to each illustration. For clarity feed pipes and vent pipes common to all systems are omitted from the diagrams. The term radiator may be taken to include any type of heat emitter except low-temperature embedded panels, which are dealt with separately later.

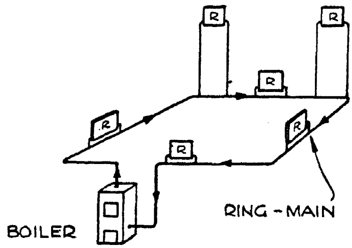


FIG. 105.—One-Pipe Ring Main System.

Fig. 105—Suitable for small installations only, otherwise the main becomes unduly large. Tends to be sluggish, particularly at the last radiators, as these are receiving water cooled by all those nearer the boiler.

With a pump this system is economical, and is often used in schools, halls and other long single-storey buildings.

Fig. 106—Generally the most economical arrangement for a multi-storey building with gravity circulation. Advantage is taken of the increased circulation produced with the radiation at high level, so enabling smaller pipes to be used than with Fig. 105. The bottom radiators are necessarily cooler than the top, and this is sometimes a disadvantage.

May be used with a pump, but Fig. 107 or 109 is to be preferred.

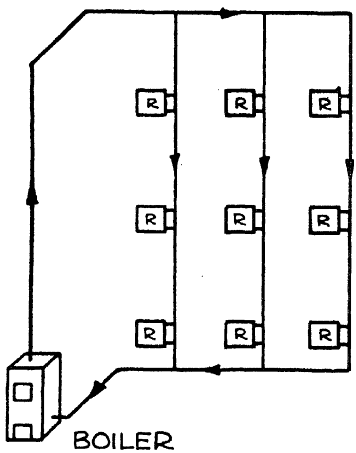


FIG. 106.—One-Pipe Drop System.

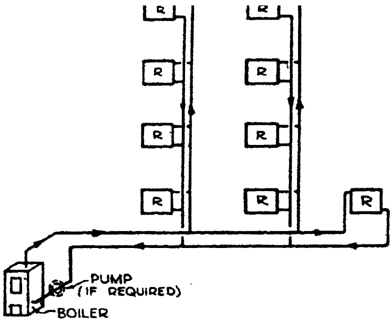


FIG. 107.—Two-Pipe Rising System.

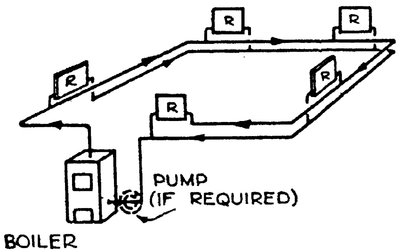


FIG. 108.—Two-Pipe System, Equalized Flow.

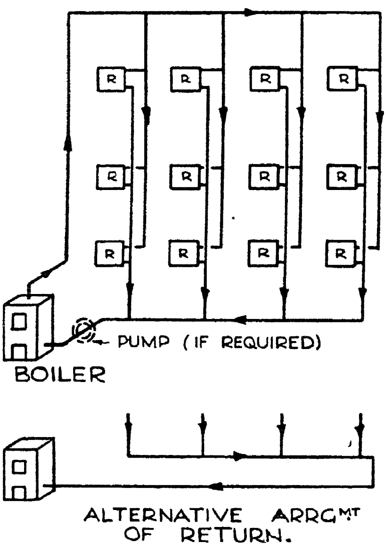


FIG. 109.—Two-Pipe Main Drop System.

Fig. 107—Possesses the great advantage that each radiator receives water at the same temperature (except for the slight loss in mains). Thus it is very responsive to boiler control. Pipe sizes are generally smaller than with the previous systems except near the boiler, but due to the double run of vertical pipes the system is not always so economical.

With a pump, this is the most satisfactory system, as each radiator circulation is accelerated thereby and smaller pipes are possible throughout; radiators may also be served below the level of the boiler.

An incidental advantage is that all large mains are kept in the basement or at ground floor, where their heat loss may be useful, as distinct from Fig. 106, which has large mains in the roof where their emission may be a complete waste.

Fig. 108—This shows the same system arranged so that the distance from the boiler to every radiator and back is the same in all cases. Often applicable to a building arranged round a quadrangle or similar case. Gives very even flow, but not generally so economical as a branched system.

Fig. 109—This is a mixture of Figs. 106 and 107, and has the advantage of equalizing the flow to all the radiators on each drop. If the return main is carried back as shown in the alternative arrangement beneath, equalized flow is given throughout the system as with Fig. 108. Is generally neither so economical nor convenient in arrangement as Fig. 107, and has

the disadvantage of large mains in or near the roof as with Fig. 106. Is equally suitable with pump or gravity circulation.

Fig. 110a—This is a cheap layout for a multi-storey block, but there is often some difficulty in accommodating the horizontal pipes at floor level, and in avoiding piers and obstructions. It is not satisfactory without a pump.

Fig. 110b—Any gravity system having radiation at or below boiler level comes under the category of 'irregular'. The system here shown is in effect a version of Fig. 105, but Fig. 106 might equally be arranged with the lowest radiators at the level of the boiler and would then be irregular. The circulation here depends on the loss of heat at the higher level causing sufficient difference in weight of the falling and rising columns to lift the cool return 'Z' back to the boiler. This is the only method by which gravity circulation may be made to operate, for example, in a private house with no basement. Circulation is generally poor if radiators R_1 are off and R_2 left on. It is equally workable with a pump as is Fig. 105.

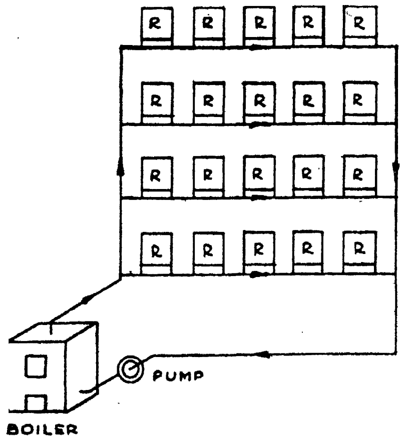


FIG. 110A.—Ladder System Pump Circulation.

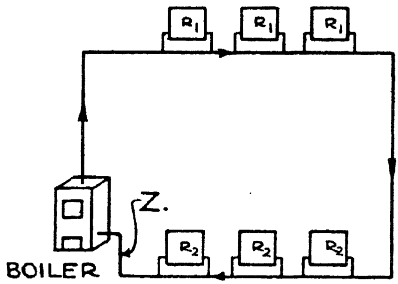


FIG. 110B.—Irregular System.

PIPE SIZING WITH GRAVITY CIRCULATION

Circulating Pressure—Fig. 111 (p. 192) shows a gravity circulation in its simplest form. Boiler A supplies heated water to radiator B at a higher level through pipes C and D which are here assumed to have no heat loss. The circulating motive force is due to the difference in weight of column H at temperature t_2 and column H at temperature t_1.

If D_1 and D_2 are the density of water at temperatures t_1 and t_2 respectively, then the difference of weight of the two columns is:

which is called the circulating pressure (CP).

Obviously the greater H is, the more circulating pressure will be available. For unit height,

$$CP = D_2 - D_1.$$

If D_1 and D_2 are in lbs. per cu. ft. then the above expression gives the circulating pressure per foot of height in lbs. per sq. ft. For purposes of pipe sizing we require this expressed in inches of water column.

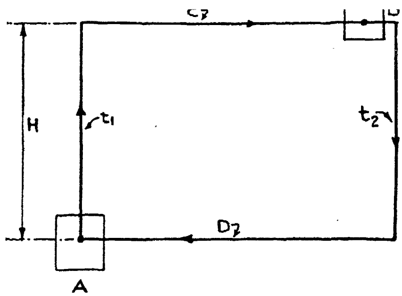


FIG. 111.—Simple Gravity Circulation.

Since 1 in. water column $= \dfrac{D}{12}$, and D at 62° F. (British standard temperature for specific gravities) $= 62{\cdot}355$ lbs. per cu. 1

then $$\frac{D}{12} = \frac{62{\cdot}355}{12}:$$

and $$\text{C.P./foot} = \frac{D_2 - D_1}{5{\cdot}196} \text{ inches W.G.}$$

Table XXXVII gives the density of water at various temperatures, from which these values may be calculated for any particular case, but for convenience a further table, No. XXXVIII, based on the formula is given, from which the circulating pressure may be read direct for the range of temperatures normally encountered in practice.

An example will illustrate its use.

In Fig. 111 height from centre of boiler to centre of radiator = 10 ft.
Assume flow 180° F. Return 140° F.
From Table XXXVIII (see p. 194) CP per ft. $= {\cdot}160$ in. water column.

$\therefore$ C.P. = 1. W.G.

Resistance to Flow—The circulating pressure assessed in the manner described is the means of creating and maintaining a circulation through the system, and if this is a closed circuit, as in Fig. 111, it will cause just such a velocity that its force is balanced by the resistance or friction encountered in the pipes, boiler and radiator.

For a given quantity of heat to be transmitted from boiler A to radiator B, water must flow through the system, the weight depending on the tem-

TABLE XXXVII

DENSITY OF WATER AT VARIOUS TEMPERATURES

Temp. °F.	Density, lbs./cu. ft.	Temp. °F.	Density, lbs./cu. ft.	Temp. °F.	Density, lbs./cu ft.	Temp. °F.	Density, lbs./cu. ft.	Temp. °F.	Density, lbs./cu. ft.	Temp. °F.	Density, lbs./cu. ft.
32	62·418	109	61·903	132	61·535	155	61·097	178	60·606	201	60·055
35	62·422	110	61·890	133	61·517	156	61·077	179	60·583	202	60·030
40	62·425	111	61·874	134	61·499	157	61·057	180	60·560	203	60·003
45	62·422	112	61·860	135	61·481	158	61·038	181	60·535	204	59·978
50	62·403	113	61·844	136	61·462	159	61·018	182	60·511	205	59·950
55	62·390	114	61·829	137	61·444	160	60·998	183	60·487	206	59·925
60	62·372	115	61·813	138	61·426	161	60·978	184	60·464	207	59·900
65	62·344	116	61·798	139	61·407	162	60·955	185	60·441	208	59·873
70	62·313	117	61·781	140	61·388	163	60·935	186	60·416	209	59·847
75	62·275	118	61·766	141	61·370	164	60·914	187	60·393	210	59·820
80	62·232	119	61·750	142	61·351	165	60·893	188	60·370	211	59·820
85	62·182	120	61·734	143	61·331	166	60·871	189	60·347	212	59·816
90	62·133	121	61·717	144	61·312	167	60·850	190	60·324	230	59·37
95	62·084	122	61·701	145	61·291	168	60·827	191	60·300	250	58·79
100	62·031	123	61·685	146	61·274	169	60·805	192	60·276	270	58·22
101	62·017	124	61·670	147	61·254	170	60·783	193	60·251	290	57·59
102	62·002	125	61·655	148	61·237	171	60·762	194	60·228	298	57·32
103	61·989	126	61·639	149	61·218	172	60·741	195	60·203	338	56·14
104	61·975	127	61·622	150	61·201	173	60·720	196	60·180	366	55·14
105	61·989	128	61·605	151	61·180	174	60·698	197	60·154	390	54·04
106	61·946	129	61·588	152	61·159	175	60·673	198	60·130		
107	61·932	130	61·571	153	61·138	176	60·652	199	60·105		
108	61·918	131	61·552	154	61·118	177	60·630	200	60·081		

perature drop. For a high temperature drop t_1 to t_2 each pound of water will carry more heat and, therefore, less water will be required than at a lower temperature. At the same time, the greater the difference between t_1 and t_2 the greater will be the circulating pressure, and obviously the greater the flow of water.

With a constant heat output from the boiler these effects strike a balance in such a way that the temperatures t_1 and t_2 adjust themselves to produce just that circulating pressure which will be absorbed in impelling through the circuit a quantity of water to an amount equal to the heat units delivered, divided by the temperature difference $t_2 - t_1$.

TABLE XXXVIII

CIRCULATING PRESSURE IN INCHES W.G. PER FOOT OF HEIGHT FOR VARIOUS TEMPERATURES

Temperature Drop Flow-Return in ° F.	Flow Temperature in ° F.													
	200°	195°	190°	185°	180°	175°	170°		160°	155°	150°	145°	140'	
	·047		·045		·043	·042		·040	·039	·038	·037	·036	·035	·034
	·070	·068	·067		·064	·062		·059	·057	·056	·054	·053	·051	·050
20°	·092	·090	·088	·086	·084	·082	·080	·078	·075	·073	·071	·069	·067	·065
25°	·114	·112	·109	·106	·104	·101	·098	·096	·093	·090	·087	·085	·082	·079
30°	·135	·133	·130	·126	·123	·120	·116	·113		·106	·103		·097	·093
35°	·156	·153	·150	·146	·142	·138	·134	·130	·126	·122	·118	·115	·111	·107
40°	·177	·173	·169	·164	·160	·156	·151	·146	·142	·137	·133	·129	·124	·120
45°	·197	·192	·187	·182	·177	·172	·167	·162		·152	·147	·141	·134	·127
50°	·215		·205		·195	·189		·178	·172	·166	·160	·152	·144	
55°	·233	·228	·223	·217	·211	·204		·192	·185	·178	·171	·163	·154	·144
60°	·251	·246	·240	·233	·226	·219		·205	·198	·190	·181	·172	·163	·153

Thus the circulating pressure balances the sum of the resistances, or in other words:

where ΣR is the sum of all the resistances to flow of water throughout the circuit.

Of these, the most important is the resistance of the piping, bends, tees, and valves. The lesser resistances are those in the boiler and radiator.

Table XXXIX (pp. 196 and 197) gives the resistance of various sizes of piping to flow of hot water.

The range of friction loss per foot therein given will be found to cover both gravity and pump circulation.

The limit to the flow of water given in the table is set by the velocity at which noise in the pipes is supposed to commence.

Local resistances, such as bends, boiler, etc., are given a factor R, according to their type, and the smaller table gives the equivalent feet run of pipe at different velocities for each resistance $R = 1$. This method is convenient in practice, though not, perhaps, strictly accurate.

Further notes on flow of water in pipes are given on p. 221.

Available CP per Ft. Run—In the first approximation of pipe sizing it is necessary to determine the total feet run or travel 'T' of the circulation, including an allowance for the single resistances, which must therefore be assumed beforehand, both for size and velocity. In the case of a branched system we take the travel for which $\frac{CP}{T}$ is the least, and this is called the index circulation.

Flow and Return Temperatures—It is first necessary to establish the flow and return temperatures, and these, in turn, depend on the mean temperature assumed in calculating the area of radiation.

With a typical radiator system this might be 160° F.

The temperature drop through the system, as already explained, determines the amount of water flowing. If the drop is too great, the last radiators will be too cool with a single pipe system, and regulation of valves will be critical and troublesome with a two-pipe system. On the other hand, too low a drop will call for inconveniently large piping and high mains losses.

A compromise is therefore generally struck at about 40° drop for gravity operation; 50° is definitely too great; 30° tends to be extravagant. Some designers take 35° as a base.

Taking 40° drop and 160° mean, the flow will be 180° and return 140°. These are common temperatures for a radiator system.

It is unusual to start with a temperature higher than 180° (though 200° would be more economical), because it would approach uncomfortably near the temperature at which the water would be converted into steam at the top of the circulating system.

Centre of Boiler and Radiator—In measuring the height of the column producing circulation it is convenient to assume the boiler to be concentrated at one central point. This is the mid-point between flow and return connections.

Similarly, the centre of a radiator should be taken at the mid-point between flow and return.

Examples follow to show how pipe sizes may be determined for gravity circulation, making use of the foregoing data.

Example (a)—One Pipe Ring Main—(See Fig. 112).

The radiators are considered as one, but, as the local circuits from flow main to radiator and back to flow main do not assist the main circulation, their height must not be taken into account in arriving at the head, which is thus the height from centre of boiler to centre of ring main.

Thus, for a temperature drop of 180° − 140° = 40°, we have (from Table XXXVIII)

or, in our case, where

ft.

$$CP = 5 \times \cdot 160 = \cdot 80 \text{ in.} \quad \ldots .(1)$$

The travel T is made up of an actual pipe length of 150 ft., plus some bend and boiler resistances which may be estimated provisionally (and subject to subsequent correction) as follows:

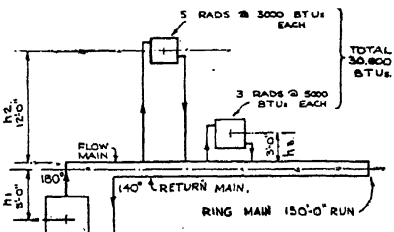


FIG. 112.—Pipe-Sizing: Example (a).

From table at bottom of Table XXXIX boiler resistance $R = 2{\cdot}5$. Let us assume provisionally $1\frac{1}{2}$ in. pipes and that there are in the main circulation 11 long radius bends, for which Table XXXIX gives us

$$R = 1 \text{ each or (for 11 bends)} = 11,$$

or with boiler, $R = 11 + 2\frac{1}{2} = 13\frac{1}{2}$, say 14.

Now assume a velocity of 1 ft. a sec. ($V = 12$ in.) when the equivalent resistance of $R = 1$ can be read from Table XXXIX (bottom) to be 5 ft. of pipe.

Hence the equivalent travel of the complete circulation is

where T_1 = actual travel, R = sum of resistances,

E = equivalent resistance, for $R = 1$, at appropriate velocity.

TABLE XXXIX

FLOW OF WATER IN PIPES at 180–140 °F

Friction Loss in Inches wg per Foot	Quantity of Water in Lbs. per Hour for Various Pipe Sizes.																		
	½"	¾"	1"	1¼"	1½"	2"	2½"	3"	3½"	4"	5"	6"	7"	8"	9"	10"	11"	12"	
·0005	—	—	—	—	270	550	1,083	1,767	2,700	3,833	6,567	10,767	16,500	24,170	38,300	44,000	56,700	72,000	
·0006	—	—	—	—	300	650	1,190	1,960	2,970	4,230	7,230	11,930	18,270	26,670	35,700	48,700	62,700	79,300	
·0007	—	—	—	200	323	703	1,300	2,130	3,227	4,600	7,870	12,970	19,830	29,000	39,300	52,700	67,700	86,300	V = 6·0'/sec
·0008	—	—	—	217	337	757	1,390	2,283	3,433	4,930	8,430	13,930	21,330	31,200	42,300	56,700	73,000	92,700	
·0009	—	—	126	230	377	807	1,490	2,433	3,670	5,270	9,000	14,900	22,670	33,200	45,300	60,700	77,700	99,000	
·0010	—	—	133	243	393	860	1,583	2,583	3,900	5,570	9,530	15,730	24,000	34,700	47,700	64,000	82,300	104,000	
·0012	—	—	147	270	433	940	1,740	2,850	4,267	6,170	10,430	17,300	26,500	38,330	52,700	70,700	90,700	116,000	
·0014	—	73	160	293	473	1,033	1,890	3,100	4,667	6,670	11,400	18,900	28,830	41,670	57,300	76,700	99,000	126,000	
·0016	—	80	172	317	510	1,107	2,033	3,330	5,000	7,167	12,330	20,330	31,000	45,000	62,000	82,300	106,000	135,000	
·0018	—	85	183	333	543	1,183	2,167	3,500	5,330	7,670	13,030	21,600	33,000	47,700	65,900	87,700	113,300	144,000	
·0020	—	89	193	357	570	1,253	2,300	3,600	5,667	8,070	13,870	22,800	34,670	50,700	70,000	93,000	120,000	152,700	
·0023	—	97	210	383	617	1,343	2,483	4,000	6,130	8,700	14,930	24,700	37,300	54,700	75,700	100,000	129,000	166,000	
·0026	—	103	223	410	667	1,440	2,650	4,300	6,500	9,330	16,000	26,400	40,000	58,700	80,700	107,300	138,300	176,700	V = 12·0'
·0030	—	112	243	443	723	1,567	2,867	4,670	7,070	10,100	17,330	28,700	43,300	63,300	87,300	116,700	150,000	190,000	
·0035	35	122	263	483	777	1,700	3,100	5,070	7,670	11,000	18,730	31,000	47,000	69,000	94,700	126,000	162,300	206,700	
·0040	40	131	283	520	837	1,833	3,330	5,430	8,270	11,870	20,170	33,300	50,700	74,000	102,300	135,300	175,000	222,300	
·0045	45	140	303	553	893	1,950	3,550	5,830	8,800	12,670	21,500	35,300	54,000	79,000	109,300	144,000	186,000	236,700	
·0050	49	147	320	583	950	2,067	3,750	6,167	9,330	13,380	22,670	37,300	57,300	83,300	115,700	153,300	196,700	250,700	
·0060	54	163	350	647	1,033	2,273	4,100	6,800	10,300	14,670	25,070	41,000	63,300	92,700	127,700	169,000	216,700	275,700	
·0070	59	177	383	703	1,133	2,473	4,500	7,370	11,170	16,000	27,230	44,700	68,700	100,000	139,000	183,300	235,700	300,000	
·0080	63	190	417	753	1,230	2,670	4,830	8,000	12,000	17,330	29,330	48,300	73,700	108,300	150,000	196,700	253,300	323,300	
·0090	67	203	437	807	1,300	2,830	5,170	8,500	12,830	18,330	31,170	51,300	79,000	115,700	160,000	210,000	270,000	340,000	
·010	71	216	467	850	1,383	3,000	5,500	9,000	13,630	19,500	32,830	54,300	83,300	122,300	165,000	221,700	286,700	360,000	
·012	79	233	513	940	1,530	3,317	6,000	9,870	15,000	21,500	36,000	59,700	91,000	132,700	182,700	246,700	316,700	400,000	
·014	85	260	557	1,017	1,660	3,570	6,530	10,730	16,330	23,900	39,300	65,000	99,300	143,300	198,300	266,700	340,000	433,300	
·016	92	277	600	1,100	1,783	3,680	7,000	11,670	17,500	25,000	42,330	69,700	106,000	154,600	213,300	286,700	360,000	466,700	
·018	97	297	637	1,167	1,900	4,080	7,500	12,330	18,670	26,700	45,000	74,000	113,300	165,000	226,700	306,700	330,000	493,300	
·020	103	314	673	1,243	2,000	4,330	8,000	13,000	19,670	28,300	47,670	78,300	120,000	175,000	240,300	323,300	410,000	525,000	V = 36·0'/sec.
·023	112	337	730	1,333	2,167	4,670	8,530	14,000	21,270	30,670	52,000	84,000	129,300	188,300	260,000	343,300	443,000	567,000	
·026	120	358	780	1,430	2,300	5,000	9,070	15,000	22,670	32,000	55,700	90,000	138,300	200,000	276,700	366,700	473,000	607,000	
·030	129	390	843	1,560	2,530	5,417	9,870	16,330	24,630	34,300	60,000	97,700	150,000	216,700	300,000	393,000	510,000	650,000	
·035	140	423	917	1,677	2,673	5,830	10,730	17,670	26,300	37,300	64,700	106,700	163,300	236,700	325,000	426,700	557,000	707,000	

V = 2·4'/sec

·040	150	453	387	1,807	2,930	6,330	11,570	18,670	28,300	40,300	69,700	115,000	175,000	265,000	384,300	460,000	597,000	760,000
·045	161	483	1,050	1,930	3,100	6,670	12,330	20,000	30,000	42,700	73,300	121,700	186,700	270,300	370,000	493,000	633,000	810,000
·050	170	513	1,116	2,033	3,270	7,100	12,830	21,000	31,700	45,300	78,300	128,300	196,700	286,700	392,000	523,000	673,000	657,000
·060	187	567	1,233	2,210	3,600	7,770	14,170	23,330	35,000	50,000	86,700	141,700	216,700	316,700	427,000	573,000	740,000	940,000
·070	203	617	1,333	2,407	3,920	8,500	15,530	25,330	38,000	55,700	93,300	164,000	235,700	350,000	468,000	627,000	607,000	1,023,000
·080	220	667	1,433	2,600	4,200	9,070	16,670	27,170	40,700	58,700	100,700	166,000	253,300	366,700	503,000	667,000	867,000	1,100,000
·090	233	703	1,533	2,750	4,500	9,700	17,670	29,000	43,700	62,300	108,300	176,700	270,000	390,000	540,000	717,000	927,000	1,170,000
·10	247	740	1,633	2,930	4,750	10,300	18,830	30,670	46,000	66,700	114,000	186,700	286,700	416,700	567,000	767,000	977,000	1,250,000
·12	277	827	1,767	3,230	5,250	11,330	20,730	34,000	51,000	73,300	126,700	206,700	316,700	456,700	626,000	833,000	1,083,300	1,367,000
·14	300	893	1,927	3,500	5,700	12,330	22,500	36,700	55,700	80,000	136,700	223,300	350,000	493,300	677,000	903,000	1,177,000	1,500,000
·16	323	967	2,070	3,770	6,170	13,270	24,330	39,300	60,000	85,000	146,700	240,000	366,700	533,300	733,000	943,000	1,267,000	1,603,000
·18	343	1,020	2,200	4,000	6,570	14,100	25,930	42,300	64,000	90,700	156,700	256,700	390,000	566,700	790,000	1,033,000	1,347,000	1,707,000
·20	363	1,077	2,330	4,230	6,930	15,000	27,330	44,300	67,300	96,700	165,000	272,700	413,300	600,000	827,000	1,100,000	1,420,000	
·23	393	1,167	2,530	4,570	7,500	16,070	29,670	48,000	73,300	103,300	178,300	292,300	443,300	650,000	883,000	1,183,000		
·26	420	1,250	2,700	4,900	8,000	17,330	31,670	52,500	79,300	111,000	190,000	313,300	476,700	693,300	950,000			
·30	453	1,350	2,920	5,270	8,670	18,670	34,000	55,700	84,000	120,000	206,700	333,300	516,700	750,000				
·35	490	1,467	3,170	5,750	9,330	20,330	37,000	60,300	90,700	130,000	223,300	366,700	556,700					
·40	527	1,573	3,400	6,170	10,000	21,670	40,000	65,000	98,000	140,000	240,000	391,700	600,000					
·45	560	1,683	3,600	6,600	10,670	23,170	42,300	69,000	104,000	150,000	256,700	416,700						
·50	600	1,773	3,830	7,000	11,330	24,530	45,000	73,300	110,700	159,300	270,000	441,700						
·60	657	1,967	4,200	7,670	12,570	27,000	49,300	80,700	122,700	175,000	300,000							
·70	713	2,133	4,570	8,330	13,670	29,500	54,000	87,300	132,700	190,000								
·80	767	2,283	4,900	9,000	14,670	31,700	57,700	93,800	142,300									
·90	813	2,433	5,230	9,530	15,670	33,300	61,700	100,000										
1·00	870	2,600	5,570	10,030	16,670	35,800	65,000	106,700										
1·20	967	2,870	6,170	11,270	18,330	39,300	72,700											
1·40	1,037	3,127	6,670	12,230	20,000	42,700	78,700											
1·60	1,100	3,330	7,200	13,170	21,400	46,000												
1·80	1,177	3,570	7,670	14,100	22,830	49,200												
2·00	1,267	3,770	8,130	14,800	24,330													
2·30	1,367	4,070	8,730	16,030	26,330													
2·60	1,467	4,330	9,830	17,170														
3·00	1,577	4,670	10,070	18,670														
3·50	1,717	5,100	11,000															
4·00	1,840	5,400	11,830															

V = 6·0"/sec.

V = 12·0"/sec.

V = 36·0"/sec.

V = 72·0"/sec.

V = 120·0"/sec.

V = 72·0"/sec.

V = 120·0"/s.

Local Resistances

Gate Valve	R = 0·5	Boiler	R = 2·5
Globe Valve	R = 2·0	Radiator	R = 3·0
Bend, Short Radius	R = 2·0	Tee, Straightway	R = 1·0
Bend, Long Radius	R = 1·0	Tee, Branch	R = 1·5
Angle Valve	R = 4·0	Enlargement	R = 1·0

Circulating Velocity in Ins. per Second	Nominal Bore of Pipe in Inches ½"	¾"	1"	1¼"	1½"	2"	2½"	3"	3½"	4	5"	6"	7"	8"	9"	10"	11"	12"
2·4"	1·0	1·5	2·5	3·0	4·0	6·0	8·0	10	12	14	18	24	—	—	—	—	—	—
6·0"	1·2	2·0	3·0	4·0	4·5	7·0	9·0	11	14	16	20	26	30	37	41	53	60	64
12·0"	1·3	2·0	3·0	4·0	5·0	7·5	10	13	15	18	22	30	34	41	47	57	66	71
36·0"	1·6	2·3	4·0	5·0	6·0	8·0	11	14	17	20	23	32	38	46	55	63	72	80
72·0"	1·75	3·0	4·0	5·5	7·0	9·5	13	16	19	25	28	36	43	54	65	72	81	93
120·0"	2·0	4·0	4·5	6·0	8·0	10·0	14	18	22	26	30	39	45	60	70	80	90	100

Equivalent Resistance of R = 1 in Feet of Pipe of Various Sizes.

In our case $$T = 150 + 14 \times 5 = 220 \text{ ft.} \qquad (2)$$

Whence from equations (1) and (2)

$$\frac{CP}{T} = \frac{\cdot 80}{220} = \cdot 0036 \text{ in. per ft.} \qquad (3)$$

The heat loss from the ring main must be allowed for, but this can only be done by estimating its size in advance. Assuming it again to be $1\frac{1}{2}$ in. unlagged at 100° difference (i.e. between water temperature and room temperature), its emission (Table XXXIII) is 110 B.T.U.'s per ft. × 150 ft. = 16,500 B.T.U.'s. This, added to the radiator emission, gives the total heat to be transmitted through the ring main thus:

Total heat transmitted = radiator emission + main losses
= 30,000 + 16,500
= 46,500 B.T.U.'s per hour.

If the temperature drop is 40°, it is clear that each lb. of water transmits 40 B.T.U.'s. Hence water required to transmit 46,500 B.T.U.'s is

$$\frac{46{,}500}{40} = 1162 \text{ lbs. per hour.}$$

From Table XXXIX, taking resistance at ·0036 in. per ft. [see equation (3)] a $1\frac{1}{2}$ in. pipe passes 800 lbs. of water per hour, a 2 in. pipe 1750 lbs.

The least commercial pipe size to pass 1162 lbs. per hour is therefore 2 in.

Our previous calculations, based on an assumed $1\frac{1}{2}$ in. pipe, therefore need correction as follows:

The pipe emission is (for 2 in. pipe) 150 ft. × 135 B.T.U.'s per ft.

	= 20,250 B.T.U.'s.
Radiators as before	= 30,000 ,,
Total	50,250 ,,

∴ Water to be circulated = 40°
= 1256 lbs. per hour.

We can now check our assumed velocity of 12 in. per sec. 1256 lbs. per hour through a 2 in. pipe produce a velocity of, say, 3 in. per sec. (by interpolation on Table XXXIX).

This is (as might be expected) rather different from the 12 in. we arbitrarily assumed, and we could correct our calculations by assuming something intermediate, such as a velocity of 6 in. per sec. It is, however, now possible to arrive at the resistance more directly as follows:

A 2 in. pipe passing 1256 lbs. an hour has a resistance (interpolating on Table XXXIX) of approximately ·002 in. per ft. With 2 in. pipe, the equivalent resistance of bends + boiler (from table at bottom, remembering that velocity is 3 in. per sec.) is about $6\cdot 1 \times R$.

So that the total travel is

$$T = 150 + (14 \times 6\cdot 1) = 235\cdot 4 \text{ ft.}$$

Hence resistance of circuit is:

Equivalent travel × resistance per ft. = 235·4 × ·002
= ·47 in W.G.

This is less than the ·80 in. *CP* available, and would therefore be satisfactory. Actually the velocity will increase slightly, and the temperature drop will decrease a little, until the reduced *CP* available exactly balances the increased resistance.

In theory, a part of the circuit could be reduced to $1\frac{1}{2}$ in. pipe, but in practice the increased circulation is more valuable than the insignificant saving which could thus be effected.

The radiator connections are sized separately, as if they had a boiler at the centre of the pipe to which they are connected. A drop of 20° may be assumed for these, and the sizes should be generous, as far as practical radiator connections permit. It is clear that the last radiators will receive water at a temperature not much above the 140° return temperature (actually 147°, see later), and will thus return water to the return main at a temperature less than 140°. But mixing with water slightly above 140° it will give 140° in the return.

It has been assumed (see diagram) that all radiators come under two groups—those on first floor height h_2 and those on ground floor h_3 (Fig. 112).

	First Floor Rads.	Ground Floor Rads.
Height	$h_2 = 12$ ft.	$h_3 = 3$ ft.
CP/ft. (160-140)	$= \cdot 075$ in.	$= \cdot 075$ in.
Available CP	$= 12 \times \cdot 075$ in.	$= 3 \times \cdot 075$ in.
	$= \cdot 90$ in.	$= \cdot 225$ in.
	$T = 24 + 8R$*	$= 6 + 7R$†
(taking R at $1\frac{1}{2}$)	$= 36$	$= 16$
	$\dfrac{CP}{T} = \dfrac{\cdot 90}{36}$	$= \dfrac{\cdot 225}{16}$
	$= \cdot 025$ in.	$= \cdot 014$ in.
Rad. and connections, emission	$= 3000$ B.T.U.'s	$= 5000$ B.T.U.'s,
at 20° drop	$= 150$ lbs.	$= 250$ lbs.
Size required (Table XXXIX)	$= 1$ in.	$= 1\frac{1}{4}$ in.

In the above example the temperatures of 160° and 140° taken for flow and return will apply to the radiators beyond the half-way point on the ring. Those nearer the boiler should be sized for higher temperatures, and the corresponding sizes would then be $\frac{3}{4}$ in. for first floor and 1 in. for ground floor. Generally, it will be found that those nearer the boiler with short runs will suffice with a $\frac{3}{4}$ in. or 1 in. connection, according to the heating surface, and those towards the end 1 in. or $1\frac{1}{4}$ in. With modern radiators, $1\frac{1}{4}$ in. is the maximum tapping possible. The $\frac{1}{2}$ in. size meets the case of only the smallest radiator with gravity circulation from a single pipe.

It should be noted that with the one-pipe system, where the outlet water from the radiators is returned into the main pipe, the temperature is reduced after each connection. When the pipe sizes have been established, therefore, it is necessary to adjust the areas of the radiators accordingly. Taking the total drop at 40° as before, and assuming the emission (pipes + rads.) to each of six radiators to be equal, this may be apportioned thus:

ad.	Flow	Return	Mea
1	180	160	170
2	173	153	163
3	166	146	156
4	160	140	150
5	154	134	144
6	147	127	137

Instead of the 160° mean at first assumed, radiators 1 and 2 having a higher mean should be reduced and 3 to 6 with lower should be increased.

Connections from a single pipe ring are usually taken from the top and side of the pipe for flow and return respectively as in Fig. 113, otherwise a reversed circulation tends to take place, which may lead to erratic operation.

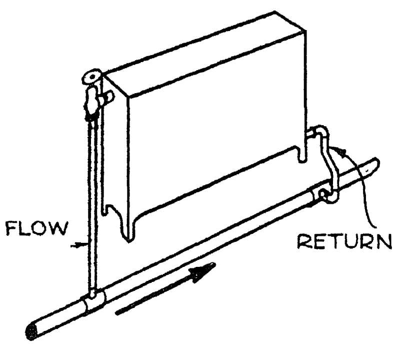


FIG. 113.—Radiator Connections.

* Assuming one radiator valve, four bends.

† Assuming one radiator valve,t hree bends.

Example (b)—One-Pipe Drop—It is necessary to consider the index circulation first; this appears as in Fig. 114. The various radiators and flow and return mains may be considered as effective at one point known as the 'average height'.

With regard to the emission of the riser and drop, the effect of these is ignored.

The emission from top and bottom mains p_1 and p_5 does not cause so great a temperature fall relative to the fall in the drop itself (where, as in this case, three drop pipes occur) as if there were

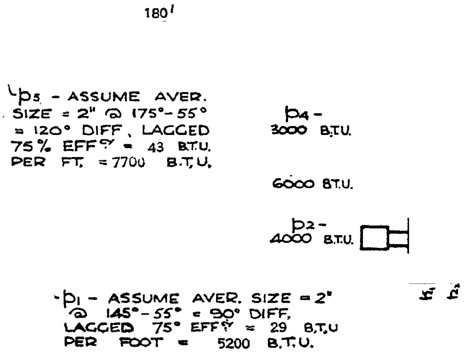


FIG. 114.—Pipe-Sizing, Example (*b*): Diagram of Index Circulation only. In all there are three drops as Fig. 115.

one only. If the emission from the three drops is roughly equal, and they are equally spaced, it will give a close approximation if the total of top and bottom mains is divided by 3.

In this case the average height is determined thus:

$$\frac{\dfrac{h_1p_1}{3}+h_2p_2+h_3p_3+h_4p_4+\dfrac{h_5p_5}{3}}{\dfrac{p_1}{3}+p_2+p_3+p_4+\dfrac{p_5}{3}}$$

$$=\frac{\left(\dfrac{4\times 5200}{3}\right)+(6\times 4000)+(16\times 6000)+(26\times 3000)+\left(\dfrac{35\times 7700}{3}\right)}{\dfrac{5200}{3}+4000+6000+3000+\dfrac{7700}{3}}$$

$$\frac{}{17{,}300}=17{\cdot}1 \text{ ft. average } H \text{ (approx.).}$$

$$CP(180^\circ-140^\circ)={\cdot}160\times 17{\cdot}1=2{\cdot}73 \text{ in.}$$

$$T=180+180+35+35+(20R \text{ at } 5 \text{ ft.}=100)=530 \text{ ft.}$$

$$\frac{CP}{T}=\frac{2{\cdot}73}{530}={\cdot}005 \text{ in. per ft. approx.}$$

From Table XXXIX a small schedule may be drawn up with ·005 in. per ft. as the resistance, thus:

Size	Lbs.	× 40 =	B.T.U.'s
$\frac{3}{4}''$	147		5,880
1″	320		12,800
$1\frac{1}{4}''$	583		23,320
$1\frac{1}{2}''$	950		38,000
2″	2067		82,680

Fig. 114 may now have the emissions filled in, taking the index drop at 20,000 total (including $\frac{1}{3}$ of mains) and the loss from the drop itself and the other two, for the sake of the example, also at 20,000. (See Fig. 115.)

The first approximation to the pipe sizes may then be filled in from the schedule in the manner shown.

Now that the sizes of the mains are known it is possible to re-calculate the emission and give each drop its correct proportion. It is also possible to calculate the resistance of the mains up to the first drop, deduct this from the *CP* for that drop, and arrive at a higher $\frac{CP}{T}$ for that pipe. Reference to the table will make it clear whether the size can be reduced to 1 in. Similarly, the second drop may also be dealt with separately.

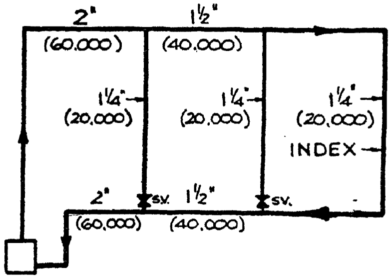


FIG. 115.—Example (*b*): see Text.

Any tendency for the first two drops to short-circuit the last, in the event of the *CP* not being entirely balanced by the resistance in each case, may be checked by the insertion of two valves marked *SV* in Fig. 115. Such valves are of the lock shield type without wheels.

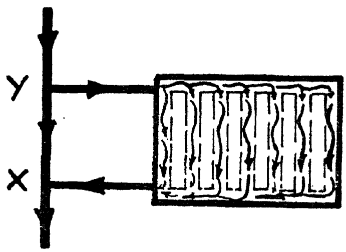


FIG. 116.

The sizing of the radiator connections from the drop pipes, as in example (*a*), requires to be calculated independently of the main pipes. Within the limits of commercial pipe-diameters, this pipe-size can only be approximate, but it is desirable to have it larger than necessary rather than too small.

The horizontal connection in Fig. 116 is sized as if there were a boiler at *X*. *CP* is established, as before, for the temperature drop available, taking this at 20°. *T* is the travel from the drop to the radiator and back, plus single resistances *R* bends and radiator.

Again, the sizes must range between $\frac{3}{4}$ in. and $1\frac{1}{4}$ in., the latter being the maximum possible with modern radiators.

When the radiator is some distance from the drop it is common practice to reduce the size of the drop from *Y* to *X* by one pipe size to augment the flow through the branch. Some designers use tongued tees for the same purpose.

Another and more positive method is to carry down the return 18 or 24 in. to act as a cool leg, in effect lowering the level of the imaginary boiler '*X*'. (See Fig. 117.)

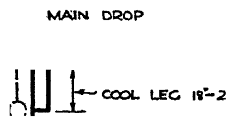


FIG. 117.

As with the single-pipe ring main the radiator surfaces should now be adjusted in accordance with the revised mean temperature, assessed as before. The top ones will be reduced, the middle will probably remain, and the bottom ones will be increased.

Example (c): Two-Pipe Rising System—Where it may be applied, this is by far the most satisfactory system for reasons already stated, but accurate balancing of the circulation is essential for success, especially if the apparatus is extensive, with a large number of branched circuits.

For this example Fig. 118 will be considered.
Index radiator obviously No. 1.
Flow and return $=180^\circ - 140^\circ$.

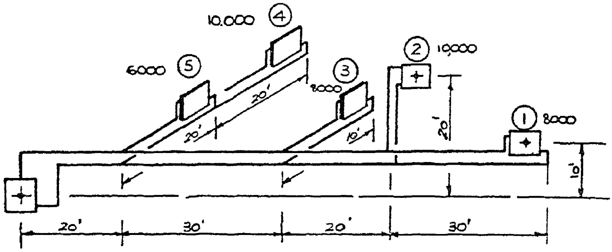


FIG. 118.—Pipe Sizing: Example (*c*).

$CP = \cdot160 \times 10 = 1\cdot60$ in.

$T = 40+60+40+60+20+20$ (piping in boiler house) $+ (15R = \text{say})\ 70$
$= 310$ ft.

$$\frac{CP}{T} = \frac{1\cdot60}{310}$$
$= \cdot0052$ in. per ft.

This gives the following table of capacities:

Size	Lb.	×40 =	B.T.U.'s
$\frac{1}{2}''$	50		2,000
$\frac{3}{4}''$	150		6,000
$1''$	330		13,200
$1\frac{1}{4}''$	600		24,000
$1\frac{1}{2}''$	980		39,200
$2''$	2130		85,200

If there were extensive circulations at upper floors instead of only the one radiator shown, similar tables for the increased *CP* and different travel '*T*' should be taken out for each.

First Approximation. For the first approximation it is necessary to allocate a proportion of the mains to each radiator by estimation.

These vary generally between 10 per cent. and 33 per cent. of the radiator emission. In this example assume 25 per cent.

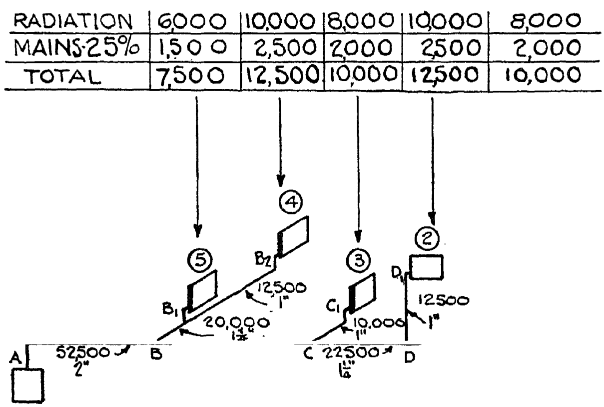


FIG. 119.—Pipe Sizing: Example (*c*).

The effect of mains loss is to cool both flow to the radiator and the return from it, so that with a given temperature difference at the boiler, the radiators have to pass more water than their actual emission calls for.

This percentage is not equal for all the radiators, but it will suffice for the first sizing.

Fig. 119 shows this addition made to each, and the totals back to the boiler.

From the table of capacities previously arrived at it is possible to insert the approximate sizes shown. Obviously the same table will not apply to the branch circuits, these having much shorter travel, but the sizes have been taken from this and will be corrected later.

Accurate Method. Now that the approximate sizes are known it is possible to calculate the heat loss from the mains section by section as follows:

Section	Size	Length	Emission per Ft. Lagged	Total
A—B	2″	80′	34	2700
B—C	$1\frac{1}{2}$″	60′	27	1000 (say)
C—D	$1\frac{1}{4}$″	40′	25	1000
D—E	1″	60′	20	1200
$B—B_1$	$1\frac{1}{4}$″	40′	25	1000
$B_1—B_2$	1″	40′	20	800
$C—C_1$	1″	20′	20	400
$D—D_1$	1″	20′	20	400
				8500

(*Note.* Total Radiation = 42,000, Mains 8500, Ratio = 20 per cent., not 25 per cent. as taken.)

These mains losses may now be allocated to each radiator in proportion to the sums of the emissions, working from the end as Fig. 120.

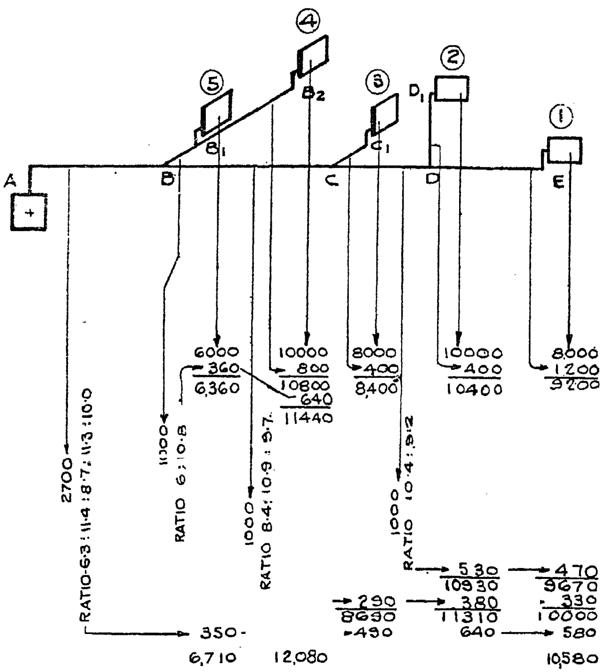


FIG. 120.—Pipe Sizing: Example (*c*).

The totals therefore appear as on Fig. 121.

A schedule is now constructed and the resistance of each section calculated. On this (see opposite page) it is possible to make corrections to the sizes, so that the actual resistance of each circuit is brought as near to the available *CP* as possible.

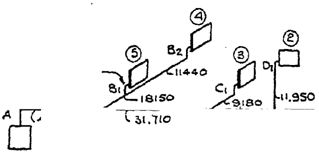


FIG. 121.—Pipe Sizing: Example (*c*).

A larger system would benefit more by accurate calculation than that taken in the above example.

Absolute accuracy of pipe sizing is in practice unattainable because of the limitation of commercial pipe sizes, the variation in their internal roughness and the fact that probably no system is ever installed exactly as designed. Further, must be remembered that the resistance allowed for bends and fittings can be no more than an approximation unless the characteristics of every item are exactly determined beforehand.

It is, however, necessary to balance each circuit against its neighbour, or short circuiting by the nearer branches will occur.

The best compromise therefore appears to be to size the pipes with fair accuracy in the manner described above, always leaving a small margin in hand to be finally adjusted by means of the lock shield regulating valves, which should be provided on every radiator.

If the piping is installed in accordance with the first approximate sizing only, the maximum economy will not be obtained and adjustment of the valves may be critical.

Example (d): Two-Pipe Drop System—From what has been said under Example (*c*) the method of approach to this problem will be apparent.

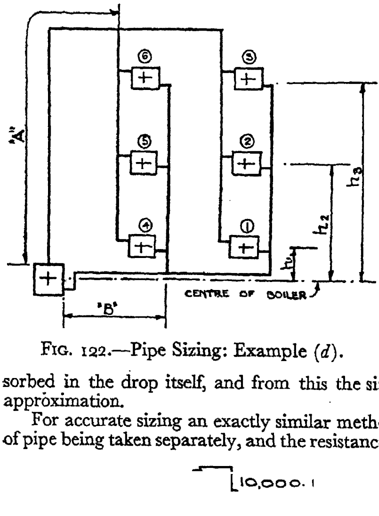


FIG. 122.—Pipe Sizing: Example (*d*).

The index *CP* in Fig. 122 will be that to radiator No. 1 calculated for h_1. The travel T will be the same for all radiators on this drop.

On the basis of this $\frac{CP}{T}$ the mains will be sized.

Radiators 2 and 3 will have greater *CP*'s calculated from heights h_2 and h_3 respectively. This will enable their flow connections and accompanying drops down to the common return to be sized for a higher $\frac{CP}{T}$.

The second drop in this example, serving radiators 4, 5, and 6, is most easily sized by deducting from the relevant *CP* the resistance of mains A and B, already sized for the index circulation. This will give the portion to be absorbed in the drop itself, and from this the sizes may be more closely estimated on the first approximation.

For accurate sizing an exactly similar method may be employed as for case (*c*), each length of pipe being taken separately, and the resistances totalled against the *CP* for each radiator.

It will be noted that no question of 'average height' arises as in the case of Example (*b*).

Example (e): Irregular System—The method of ascertaining the circulating pressure in this case is entirely different from that in the foregoing examples as the '*CP* per foot of height' method cannot be applied owing to the complication of the cool return lift back to the boiler. It is therefore necessary to return to first principles.

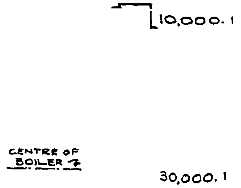


FIG. 123.—Pipe Sizing: Example (*e*).

In Fig. 123 all the first floor radiators are assumed to be concentrated in one R_1, and all those at ground floor in R_2.

The circulating pressure is that caused by the difference of weight of the columns P_1 and $N_1 + N_2$, thus:

$$CP = P_1 - (N_1 + N_2).$$

The temperatures of N_1 and N_2 are known (say 180° and 140°), that of P_1 must be calcu-

Tabulation of pipe sizes, see Example (c) opposite.

Section	Size	Length (Incl. R)	B.T.U.	Lbs. $= \frac{\text{B.T.U.}}{40}$	Res./Ft. (Table XXXIX)	Total Res. (Length × Res./Ft.)	CP available Inch Water Column
A—B	2″	~~130~~	49,860	1246	·0020	~~·26~~	Allowing for mains 2′ below Index Rad.
correction		2″ 70′ 1½″ 60′			·008	·14 ·48	
B—C	1½″	60′	31,710	793	·0037	·22	Revised $H = \frac{(10 \times 8) + (8 \times 2{\cdot}5)}{8 + 2{\cdot}5}$
C—D	1¼″	40′	22,530	563	·0047	·19	=9·5 ft.
D—E	1″	80′	10,580	264	·0035	·28	$CP = 9{\cdot}5 \times {\cdot}160$
						~~·95~~	
		[310′=T]				1·31	=1·52
						(surplus ·21)	
Branches							
B—B_1	~~1¼″~~	50′	18,150	454	~~·0038~~	~~·19~~	
Correction	1″				·0096	·48	
B_1—B_2	1″	50′	11,400	286	·004	·20	
add A—B	as	above				·62	
						1·30	1·52
						(surplus ·22)	
Radiator-Connection							
B_1	¾″	10′	6,710	158	·006	·06	
A—B_1						1·10	
						1·16	1·52
						(surplus ·36)	
C—C_1	~~1″~~ ¾″	25′	9,180	230	~~·003~~ ·012	~~·08~~ ·30	
A—C						·84	
						1·14	1·52
						(surplus ·38)	
D—D_1	~~1″~~	25′	11,950	300	~~·0045~~	~~·11~~	1·52
Correction	½″ ¾″	13′ 12′			·14 ·14	1·82 ·21	$+10 \times {\cdot}160 = 1{\cdot}60$
A—D						1·03	3·12
						3·06	
						(surplus ·06)	

lated, it being intermediate in proportion to the relative total emissions at the top and at the bottom, thus:

Top. Assume pipe is $2\frac{1}{2}$ in., mean water 170°, air 60°, difference 110°, emission 174 B.T.U.'s per ft. × 40 ft. - - - - - - - - - 6,960
Radiators - - - - - - - - - - - - 10,000

say 17,000 B.T.U.'s.

Bottom. Pipe $2\frac{1}{2}$ in., mean water 150°, air 60°, difference 90°, emission 135 B.T.U.'s per ft. × 40 ft. - - - - - - - - - - 5,400
Radiators - - - - - - - - - - - - 30,000

35,400 B.T.U.'s.

(The emission from riser and drop is ignored as before.)

Temperature of $P_1 = 180 - \Big($

$= 180 - 1$

It is now possible to evaluate

$CP = P_1 -$

by taking out the heights and densities (Table XXXVII) as follows:

$$CP = (15 \times 60{\cdot}850) - \{(12 \times 60{\cdot}560) + (3 \times 61{\cdot}388)\}$$
$$= 912{\cdot}750 - (726{\cdot}720 + 184{\cdot}164)$$
$$= 1{\cdot}866 \text{ lbs. per sq. in.}$$

Dividing by 5·196 (see p. 192), i.e. by 5 (approx.) to convert to inches water column

$CP = {\cdot}37$ in. water column.

Then $T = 12 + 40 + 15 + 40 + 3 + (10R \text{ at } 8 = 80) = 190$ ft.

Total emission = 17,000 + 35,400 - - - - - - - - - = 52,400
+ (risers and drop on 160° water = 100° diff.) 30 × 154 = 4,620

say 57,000 B.T.U.'s.

= 1425 lbs.

Reference to Table XXXIX shows that at ·0019 in. per ft.

2 in. passes 1218 lb.
$2\frac{1}{2}$ in. ,, 2233 lb.

The main must therefore be $2\frac{1}{2}$ in., though, as this is too large, a portion could be reduced to 2 in. Actually the increased size will allow a lower temperature drop than 40°, and the economy is not worth making. As it has so happened that $2\frac{1}{2}$ in. was the size chosen in the first instance in determining the emission of the main, there is no need to revise this, though it would have been necessary otherwise.

The sizes of the radiator connections are determined independently, as for example (*a*), and again it must be remembered that the radiation surfaces require adjustment on account of the different temperatures of water with which they are supplied.

It will be appreciated that the above example is the simplest possible for this class of circulation, and in practice the numbers of positive and negative legs to be considered usually make the calculation of such systems somewhat laborious, particularly if there are a number of branches, each of which, it will be found, affects the main or index circuit.

PIPE SIZING WITH PUMP CIRCULATION

It is first necessary to establish the pump head for which the system is to be designed. Economy of running cost in some cases calls for this to be kept low, but except in large systems this aspect is generally unimportant.

One method is to assume beforehand a head of 5, 10 or 15 ft. or more, and size accordingly. Another is to allow a friction loss of ·10 in. or ·15 in. water column per ft. run of pipe and let the head come what it will. This

gives a rational basis, as the head is consistently greater the larger the system. Thus, if the travel is 1000 ft. the head at $\cdot 10$ in. per ft. will be

$$1000 \times \cdot 1 = 100 \text{ in.} = 8 \text{ ft. } 4 \text{ in.}$$

Important points to notice in this connection are the following:

(1) The resistance of bends and fittings will be very much greater proportionately to that of the piping than with gravity circulation, and may be equivalent to 50 to 100 per cent. of the actual pipe run.

(2) The resistance of the pump connections is often ignored, though this may be as much as 2 ft. of the total, and must therefore be allowed for.

(3) The gravity circulating head may be ignored except in high buildings. Even here it will be found to have little effect on the pipe sizing.*

(4) Pump heads up to 70 or 80 ft. may be allowed where extensive runs and large volumes of water call for high velocities for maximum economy of pipe sizing. This is particularly so where the pump may be steam-driven and the exhaust steam used through a calorifier for heating the water, in which case the circulation costs are practically nil.

(5) The temperature drop assumed for purposes of arriving at the weight of water to be circulated should not be more than 30°, or excessively fine regulation will be called for. 20° or 25° are sometimes worked to.

Single-pipe ring-main or ladder systems are best sized for 20° drop, so as to reduce the temperature difference between the extreme radiators.

Example—The two-pipe system only will be considered, as this is by far the most advantageous with pump circulation.

As with Example (*c*) (p. 201), the emissions of the radiators should be marked on the plans or on a diagram similar to Fig. 124, and totalled back to the boiler with a suitable allowance for

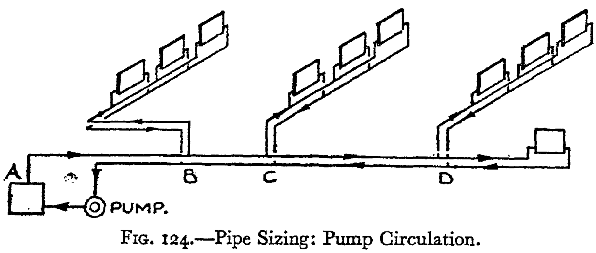


FIG. 124.—Pipe Sizing: Pump Circulation.

The travel 'T' is then measured to the most distant point, as at 'E', and an appropriate allowance made for bends and resistances by estimating the number of each from the drawings and multiplying the total by an appropriate average foot run each from Table XXXIX.

The pressure loss per foot run is then determined either arbitrarily at $\cdot 1$ in. or $\cdot 15$ in. or more per foot; or from the pump head fixed beforehand thus:

Assumed pump head - -	5
Allowance for pump connections	2 ft.
Available - - -	13 ft.

$T = 1300$ ft. (including resistances).

$$\frac{CP}{T} = \frac{13 \times 12}{1300} = \cdot 12 \text{ in. per ft.}$$

* Where heating elements are below the boiler or mains the gravity 'back pressure' should not be ignored.

From this a table may be drawn up for the first approximate pipe sizing (as with gravity circulation, Example (*c*)) and the sizes of the index circuit filled in, together with other circuits of somewhat similar travel.

For shorter circulations branching off nearer the boiler, as at *B*, time may be saved by first estimating the resistances of pipes *A* to *B* (flow and return), deducting this from the total head available, and constructing a separate table with a higher $\frac{CP}{T}$ for this branch only.

Accurate sizing may then be proceeded with on exactly similar lines to that already discussed in example (*c*). The mains emission should first be re-estimated and properly proportioned to each radiator. Following this the resistance of each section must be taken out, and each branch balanced against what is available. Lock shield regulating valves on radiators and branch mains are in any event essential with a pumped system for final regulation, as again meticulous accuracy in pipe sizing is unattainable.

Radiator connections will be found usually to be ½ in. ⅜ in. is often sufficient, but the risk of blockage is too great.

The Capacity of the Pump is determined from the total B.T.U. emission of the system by dividing this by the temperature drop assumed. The answer in pounds is then converted into gallons per minute by dividing by 60×10, this being the usual method of rating pumps.

To limit the pump to this duty would mean that every circuit and radiator is expected to take exactly its correct volume of water and no more. This is obviously impracticable, and in order to allow a margin to facilitate regulation an addition of 30 to 50 per cent. on the calculated pump duty is desirable.

In the case of single pipe or ladder systems where the water is not subdivided, or is split into some three or four circuits only, 10 per cent. margin would be sufficient.

The following example will show the application of the above:

Total emission of radiators and mains - 1,800,000 B.T.U.'s per hour.
Temperature drop - - - - 30°.

Gallons per minute $\frac{1,800,000}{30 \times 10 \times 60} = 100$ galls. per min.

A pump of, say, 150 g.p.m. would be recommended. The head would be that previously determined for the pipe sizing, rounded up to the nearest standard. No appreciable margin on the head is desirable, as the pump will tend to pass an excess of water, thus overloading the motor.

The horse power delivered to the water for such a pump, operating at, say, 10 ft. head, would be

$$\frac{150 \times 10 \times 10}{33,000} =$$

Pump efficiencies vary between 50 per cent. and 75 per cent. If the former, the horse power absorbed at the pump shaft would be

The electric motor for this duty would require to have a margin over this to allow for the possibility of error in the estimation of the head. Prob-

ably a 1·5 or 2 h.p. motor would be suitable, depending on the characteristics of the pump.

Types of pumps and their characteristics will be dealt with later.

PIPING SYSTEMS FOR EMBEDDED PANEL HEATING

Two systems only need be considered under this heading: Fig. 125, two-pipe drop; Fig. 126, two-pipe rising, which are the same in principle as Figs. 109 and 107 respectively.

Fig. 125 may be used for gravity or pump circulation. Gravity circulation of panel systems is only possible where the runs are short, as in a private house. Fig. 126 may only be used with a pump, as the circulation through the coils is reversed, i.e. water enters at the bottom and leaves at the top.

The drop system suffers from the disadvantage of large mains at, or above, roof level, and some trouble occasionally occurs with air-locking of the coils, as air and water are travelling in opposite directions. Thus it is usual to provide with such a system 'reversing connections' on the pump whereby the flow may be reversed and the water delivered up the return mains. Unless fresh water is introduced this trouble does not generally recur after the first heating season.

The riser system (Fig. 126), being constantly 'reversed', does not suffer from this difficulty nor from the objection of large mains at the top. All large mains are kept in the basement or at low level. A vent cock at the top of each riser is necessary, and requires to be opened periodically just as an air cock on a radiator is opened from time to time.

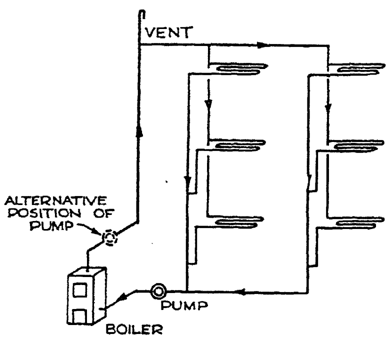


FIG. 125.—Drop System for Panel Heating.

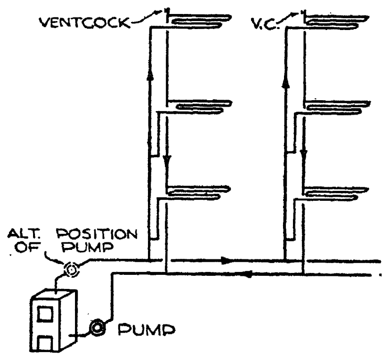


FIG. 126.—Rising System for Panel Heating.

The sizing of the pipes for panel systems is basically the same as for radiators and will not be gone into in further detail. The temperature drop normally assumed is 15°, and this means that much greater quantities of water must be circulated for the same heat output. On this account a higher friction head per foot run is called for, unless the mains are to be unduly large.

The resistance of each individual coil is, moreover, considerably greater

than that of a radiator. As much as 60 sq. ft. of panel surface may be served by one coil, which, with pipes at 6 in. centres, calls for 120 ft. run of pipe, or perhaps 135 ft. with the connections. The resistance of this length of $\frac{1}{2}$ in. bore pipe with a number of 180° bends when passing sufficient water for its own emission, plus mains losses, is about 6 ft. It will thus be found that pump heads for embedded panel systems are commonly higher than with radiators.

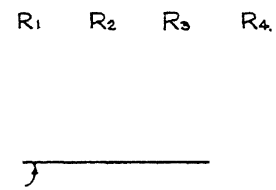


FIG. 127.—Electrical Analogy.

One important effect of the high resistance of the panel coils is the self-regulation produced thereby.

Considering an electrical analogy (Fig. 127), a number of resistance coils R_1, R_2, R_3, R_4 are supplied in parallel from common wires, the resistance of which is low compared with the coils. R_4 at the end of the run has practically the same pressure (potential difference) across its two ends, and therefore passes nearly as much current as the nearer coil R_1. Thus each coil takes its share evenly irrespective of the distance (within limits) from the point of supply.

Exactly the same conditions apply with water circulating through mains of low resistance and coils of high resistance, though as the mains in this case do in fact constitute a higher ratio than in the electrical analogy, the balancing between the various circuits is to that extent not so perfect without the aid of careful pipe sizing or regulating valves.

Another fact also emerges from the electrical comparison. If one of the coils, say R_2, is lower in resistance than the others it will pass relatively more current. In the water system this means that the shorter the coil and smaller the panel the more water will it allow to flow through. This is exactly the opposite of what is desired, but must, in fact, occur in practice, since these systems are rarely regulated to any degree of fineness. The problem might be overcome by calibrated resistance discs for each size of panel, inserted in the pipe, but this introduces risk of blockage, and so far does not appear to have been done.

The low temperature drop assumed for this type of apparatus, however, overcomes these apparent disparities. If one small coil is passing twice as much water as its larger neighbour, it may have a drop of $7\frac{1}{2}°$ as compared with 15°, but this would have little effect, since the mean might be only $3\frac{3}{4}°$ different, which would be insignificant.

It will be seen that the method of sizing on B.T.U. output for panel systems is not consistent with what happens in practice, though in the average over a number of coils it is probably near the mark. Another method, though not one yet fully developed, is to assign to each coil a definite volume of water based on tests, giving more to the small than the large, and sizing accordingly. Provided the large ones were adequately supplied, this would be both simple and satisfactory.

CIRCULATING PUMPS

It has already been explained how the capacity and head of a circulating pump are calculated. The questions of type and characteristics will now be briefly referred to.

Centrifugal Pumps—The centrifugal type of pump is most suitable for the purpose. This consists of an impeller rotating in a fixed casing of volute shape. The impeller may be of the end suction type, as in Fig. 128, or of the

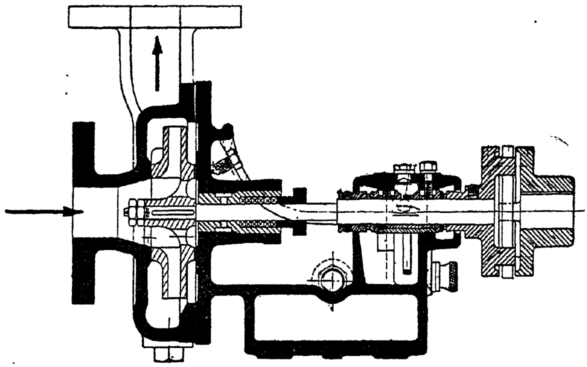


FIG. 128.—End-Suction Type of Centrifugal Pump.

split casing type as in Fig. 129, in which the top half of the casing is removable for inspection.

Centrifugal pumps of either type are more suitable for heating circulations than other types of pump since they are very conveniently and simply driven by electric motors, are low in cost, require little maintenance and may be made silent in running.

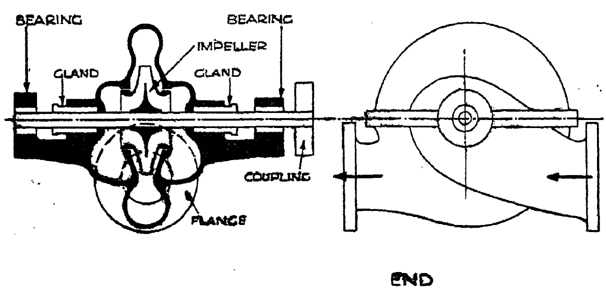


FIG. 129.—Split Casing Pump.

The end suction type is not generally so efficient as the split casing, as in the former there is an unbalanced thrust due to the water being handled by one side of the impeller only. On the other hand it is usually cheaper,

and for the smaller sizes covering most average size heating systems the slight difference in efficiency is unimportant.

Types relying on the gland to act as the only bearing are bad, as they invariably give trouble with leakage at this point in time. Those having external bearings with the gland simply acting as a water seal are to be preferred.

The closed impeller type has no out-of-balance thrust, and the two bearings on either side make it a good mechanical design. The advantage of the split casing with its freedom for easy inspection of the inside without removal of pipework is perhaps of no great value in a heating system, since sediment or clogging material is not present to give trouble, but in large installations where an annual overhaul is given this feature is useful.

Either type has characteristic curves of the form given in Fig. 130. From this it will be seen that at constant speed, the volume rises as the head is reduced. At the same time the power consumption also rises, and the efficiency rises to an optimum point and then falls off.

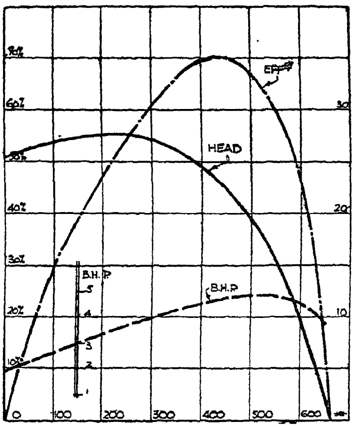


FIG. 130.
Typical Characteristics of Centrifugal Pumps.

Remembering that in a heating circulation the head is frictional, water not being actually lifted from one definite level to another, the pressure to be produced by the pump may be different from that estimated by calculation. If the head is less than the pump has been installed for, as has already been stated, the water delivered will be greater, with a higher horse-power consumption, and more current taken by the motor. This may be corrected artificially by partly closing a valve in the main circulation, but if this is not done the motor will be overloaded. Thus it is necessary to provide a motor with an ample margin of power, so that even if the head is overestimated no risk of burning-out will occur.

If, on the other hand, the frictional head is more than that for which the pump is rated, less water will be circulated and a reduced power consumption will result. The effect of this on the heating circulations will obviously be a greater temperature drop and a tendency for the more distant radiators to be cool. There is no cure for this except to speed up the pump, which is sometimes possible with a direct current motor or turbine drive, but not with alternating current.

The pump should be chosen with this point clearly in mind, and the characteristic curve will at once show which is the most suitable. Other things being equal, a volume-head curve having a flat top is better for heat-

ing circulation than one sharply peaked. The former will vary little in volume for considerable changes in head.

Pump Arrangement—Direct coupling to an electric motor is the simplest form of drive, and a flexible coupling is usual to allow for possible distortion of the base plate and misalignment of the bearings. An alternative arrangement is one in which the motor is separate, with a V-rubber belt drive to the pump. This permits the motor and pump to run at different speeds with resultant economy, and of complete mechanical isolation of the driving unit from the water circuit, with improved silence of running.

A further means of preventing the transmission of vibration from the pump to the piping is by the insertion of rubber connections in the flow and return. These are not without disadvantages, and it is questionable whether the water itself does not act as a sound transmitter just as much as the pipe, and this it is impossible to isolate. When trouble occurs due to a noisy pump, rubber connections may, however, be tried as one of the means of improvement.

Automatic By-pass—In order to allow of the circulation continuing by gravity in the event of the pump being stopped, or at night time when gravity circulation gives all the heat that is necessary, an automatic by-

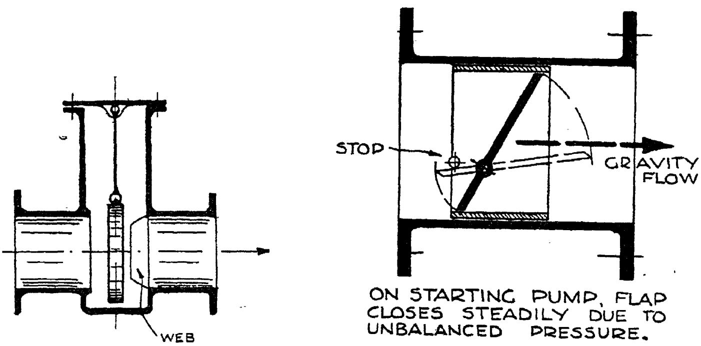


FIG. 131 (*a*) and (*b*).—Automatic By-pass Valves.

pass valve is necessary. One form of this consists of a suspended flap normally off its seating, as in Fig. 131 (*a*). On the pump being started the first tendency of the water is to flow through the by-pass in the reverse direction, so closing the valve, usually with great violence. A type devised to overcome this disadvantage and to close more slowly is indicated in Fig. 131 (*b*).

Self-Contained Pumps—Reference should be made to a class of pump developed in recent years incorporating in one unit, pump, motor, and by-pass, and in some cases including isolating valves to the pump.

Fig. 132 shows a typical example with V belt drive to the motor. A

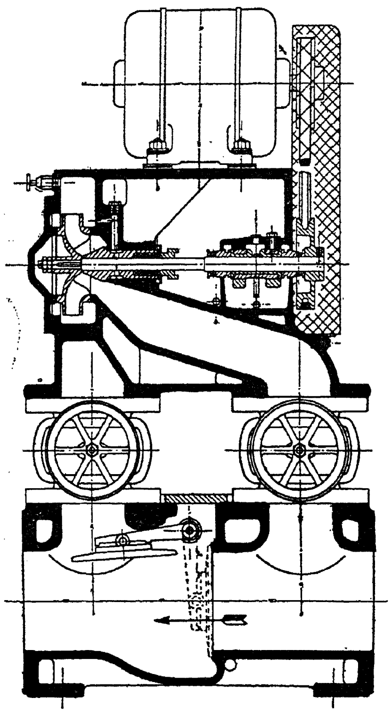


FIG. 132.—Self-contained Pump Unit. ('Criton')

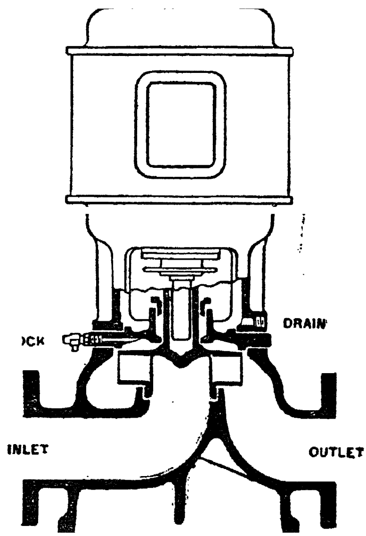


FIG. 133.—Vertical Spindle Pump. ('Mopump')

vertical spindle type (Fig. 133) is so designed as to require no by-pass, the water flowing freely through the pump when stopped. The merit of these self-contained pumps is compactness and convenience in operation.

Where a pump is relied on as the sole means of circulation, the gravity effect being too small to be useful, it is desirable to arrange for duplicate sets, with valves, so that either pump may be in commission with the other as standby.

DUAL TEMPERATURE SYSTEMS

It is sometimes necessary to supply water at two different temperatures from the same boiler plant, such as when a low temperature panel system and a radiator system occur in the same building.

This can conveniently be achieved by a mixing arrangement whereby part of the return water from the low temperature circuit is re-circulated direct into the flow without passing through the boiler. The amount of water re-circulated may be controlled by hand, or by thermostatically operated mixing valve. A separate thermometer is desirable in the low temperature flow.

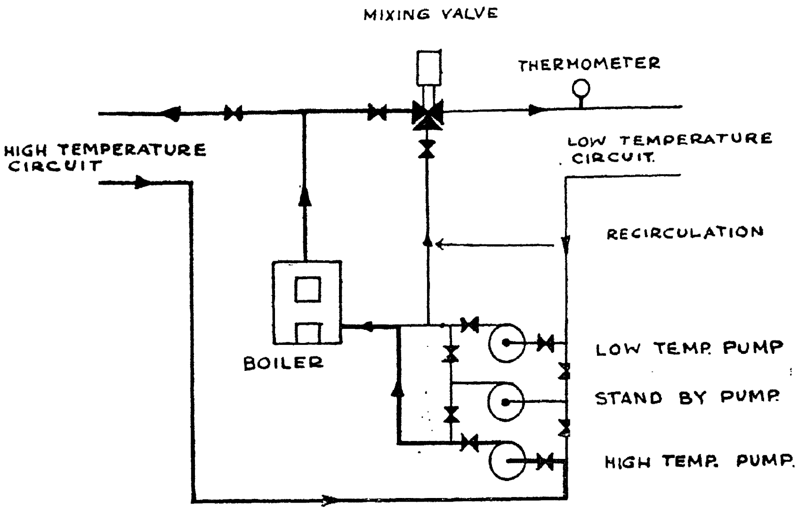


FIG. 134.—Diagram of Mixing Arrangement.

The high temperature circuit operates independently direct from the boiler in the usual way.

The arrangement is shown in Fig. 134. It will be noted that there are three pumps, one high temperature, one low temperature, and one stand-by with valves so arranged that it may serve as a duplicate to either of the other two.

HOT-WATER HEATING GENERALLY

Feed and Expansion Tanks—The water in a heating system expands on being heated, and the purpose of the feed and expansion tank is to receive this water when the system is hot and return it when it cools down. In so doing the water in the system is not changed, and encrustation or corrosion which might otherwise occur with a constantly changing supply is avoided.

For the same reason it is inadvisable to empty the system in the summer or to change the water at all, except when repairs or alterations call for it.

The expansion of water from 45° F. to 212° F. is one twenty-third of its volume at the initial temperature. In order to determine the size of expansion tank for a certain system it is therefore necessary to estimate its total water contents. For boilers and radiators, makers' catalogues may be consulted. As an approximation is all that is necessary, however, Table XL will be found to give a fair average. This table also gives the contents of piping.

In addition, about 4 in. of water is necessary permanently in the tank to float the ball valve, and a fair margin of space above, before the overflow is reached. This will call for a tank capacity about double that estimated.

TABLE XL

Contents of Heating Apparatus

(*a*) Boilers, cast-iron sectional type - 1 gall. per 6000 B.T.U.'s.
(*b*) Boiler, wrought-iron sectional type 1 gall. per 10,000–20,000 B.T.U.'s according to size.
(*c*) Radiators, 'Classic' types - - - - - 0·07 gall./sq. ft. heating
(*d*) Radiators, hospital and old plain types - - - 0·2 gall./sq. ft. heating s
(*e*) Piping:

Internal Diameter	Galls./Foot Run	Foot Run/Gall.
$\frac{1}{2}$ in.	0·0084	119·0
$\frac{3}{4}$ in.	0·019	52·63
1 in.	0·0339	29·50
$1\frac{1}{4}$ in.	0·053	18·87
$1\frac{1}{2}$ in.	0·0763	13·11
2 in.	0·1356	7·37
$2\frac{1}{2}$ in.	0·212	4·72
3 in.	0·3053	3·28
$3\frac{1}{2}$ in.	0·4156	2·41
4 in.	0·5426	1·85
5 in.	0·848	1·18
6 in.	1·221	0·82
7 in.	1·665	0·60
8 in.	2·175	0·46
9 in.	2·753	0·36
10 in.	3·398	0·29
12 in.	4·90	0·20

The method of connection of the tank to the system depends on the type of apparatus. With a simple gravity system this may be as Figs. 135 or 136.

-BOILER

Fig. 135.—Arrangement of Feed and Expansion Tank in Gravity System.

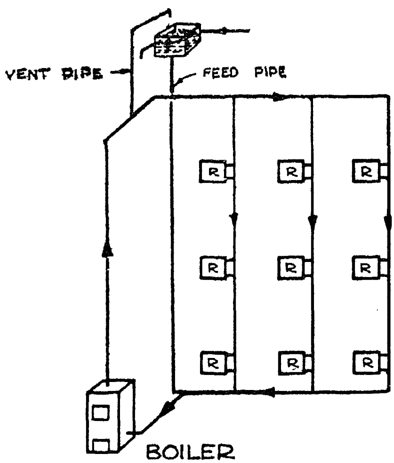


Fig. 136.—Arrangement of Feed and Expansion Tank in Gravity System.

When a pump is installed the feed-pipe is often connected to the suction as in Fig. 137.

When open vent pipes are provided on the boilers the latter method calls for the vents to be run up unduly high to prevent their discharging water, when the pump is running. For this reason a connection direct to

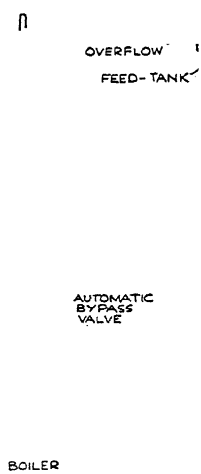


FIG. 137.—Two-Pipe System with Pump, showing Feed Tank, Feed and Expansion Pipe and Vent.

FIG. 138.—System similar to Fig. 137, but with Pump in Boiler Flow.

the boiler return is used, as in Fig. 79 (p. 142), which shows a battery of three boilers so connected, each being provided with a stop cock in the feed-pipe to permit of individual emptying for repair. Such cocks should be locked open with a padlock to prevent accidental closing, when the boiler fire is alight.

Air venting may be troublesome with high level radiators with the last method when the pump is running, and a better method is to put the pump in the boiler flow instead of in the return, as in Fig. 138.

The feed and expansion tank is usually made of galvanised iron and is provided with a ball-cock connected to the cold-water supply. The lever of the ball-cock is bent so as to keep the ball near to the bottom, and the valve itself is above overflow level. The overflow should be of large dimensions such as 1½ in. for small systems and 2 in. or more for large ones. If in an exposed position, or in a cold roof-space, the tank and its connections should be protected against frost.

Insulation of Pipes—Pipe insulation involves the question of maximum economical thickness due to the fact that the outer radiating surface is increased by the lagging.

Table XLI makes this clear. A 4 in. pipe insulated 1½ in. thick with the material in question transmits 10 per cent. of bare pipe loss and is then said to have an insulating efficiency of 90 per cent. A 1 in. pipe with exactly similar lagging has an efficiency of 81 per cent.

The table is based on authoritative tests for the materials, thicknesses and temperatures in question, carried out on a standard 4 in. diameter pipe. Values for the other sizes have been calculated from this basis.

The corresponding efficiency of various other insulating materials in

TABLE XLI.—INSULATION TRANSMISSIONS AND EFFICIENCIES

Based on 4 in. Dia. Pipe, Insulated with Glass Silk. Figures in B.T.U./Hr. 100° Difference (165° F. to 65° F.)

Pipe Dia.	Loss per F‘ Run Bare Pipe	2" Thick Loss/Ft. Run	2" Thick Efficiency	' Thick Loss/Ft. Run	' Thick Efficiency	1" Thick Loss/Ft. Run	1" Thick Efficiency	½" Thick Loss/Ft. Run	½" Thick Efficiency.
6"		23·4	93	29·3	91½%	39·5		61·5	
5"	282	21·3	½%	26·2	91%	35·0		53·3	
4"	232	18·6	92%	23·2	90%	30·2		45·2	
3"	188	17·1	91%	20·6	89%	26·5		38·0	
2½"	154	16·0	89½%	19·0	88%	24·0		34·0	
2"	129	14·6	88½%	17·1	87%	21·0		28·5	
	110	14·0	87%	16·0	85½%	19·8		25·0	
			86%	15·5		18·6		23·5	
			83½%	15·1		17·5		21·4	
¾"			81½%	14·6	79½%	16·5		19·7	
½"	56	13·0	77%	14·2	75%	15·8		18·0	

RELATIVE INSULATING EFFICIENCIES OF VARIOUS MATERIALS

(Based on 4 in. dia. Pipe, Air 65°, Mean Water 165° F.)

Material	Total Thickness 2"	1½"	1"	½"
Glass silk	92%	90%	87%	81½%
Sectional 85% magnesia	87%	85%	82%	77%
Plastic 85% magnesia supercoated ½" hard setting composition	82%	78%	73%	—
Blue asbestos sectional	82%	80%	77%	72%
Fossil meal and asbestos plastic composition	67%	62%	57%	—
Hair felt	—	—	60%	55%

common use, referred to a 4 in. pipe, are given at the bottom of the table. Values for these materials for the other pipe sizes may be obtained by proportion from the table above.

It must be remembered that the texture and density of different samples vary widely in practice and test results vary similarly. Thus the values given in Table XLI must not be taken as anything more than a guide when relative costs and thicknesses are under consideration. The figures should not be used when estimating mains losses in sizing the mains or boiler as they may not be obtained in practice. A sufficiently accurate method for this purpose is to allow an overall efficiency for insulated pipes of 75 per cent., which means that the bare pipe loss is simply divided by four.

It must also be remembered that insulation efficiencies are often given by makers on the basis of steam pipe temperatures. These may be misleading where water temperatures are concerned, as the lower the temperature difference, air to pipe, the lower may be the efficiency of any given insulation. Efficiency figures are useless unless it is known on what temperatures they are based.

As to the materials themselves, dry or sectional laggings are usually dearer than the plastic but avoid the mess, steam and smell associated with

the latter. They may be applied when the pipes are cold, whereas plastic requires heat for drying out. All materials may be finished in a variety of ways with enamels, canvas, bituminous paint, metallic sheathing, etc., according to position and cost.

Expansion of Pipes—The following table gives the increase in length of 100 ft. of steel piping when heated to the temperatures named from an initial temperature of 40° F.

	Increase for 100 Ft.
120° F. - - - -	·64 in.
160° F. - - - -	·96 in.
215° F. - - - -	1·41 in.
250° F. - - - -	1·69 in.

In practice with L.P. hot-water heating, temperatures above 200° seldom obtain, so that it is safe to approximate 1 in. expansion per 100 ft. run for the normal case. This expansion must be allowed for in the fixing of the pipes; in the case of large installations with long straight runs, by the provision of expansion joints of the sliding type, or of the loop or horseshoe type (see Fig. 139). It is frequently possible to avoid the use of expansion joints or loops by utilising changes of direction, bends, offsets, etc. In any case it is desirable to anchor the piping at definite points so that with successive expansions and contractions there is not a gradual creeping which would lead to fracture and leakage. Movement of the piping is assisted by long hangers or by rollers and chairs in place of rigid pipe-clips.

TABLE XLII

DIMENSIONS, ETC., OF EXPANSION JOINTS AND BENDS FOR HEATING PIPES

Nominal Diameter of Pipe	Type (*a*), fig. 139		Type (*b*), fig. 139				
	Length over Flanges	Allowance for Expansion	Max. Total Travel, including Cold Draw *	Max. Cold Draw *	Load on Anchors at Max. Cold Draw *	Dimension *X*	Dimension *Y*
					Lb.		
1″	10″	2″	0·90″	0·50″	170	1′ 6″	2′ 0″
1¼″	11″	2″	0·90″	0·50″	190	1′ 8″	2′ 2″
1½″	12″	2″	0·90″	0·50″	400	1′ 9″	2′ 3″
2″	14″	2″	1·05″	0·60″	600	2′ 0″	2′ 6″
2½″	16″	2¾″	1·05″	0·60″	1130	2′ 3″	2′ 9″
3″	19″	2¾″	1·25″	0·70″	1390	2′ 6″	3′ 0″
4″	20″	2¾″	1·45″	0·85″	1870	3′ 4″	3′ 6″
5″	22″	2¾″	1·85″	1·00″	2000	4′ 2″	4′ 2″
6″	23″	2¾″	2·10″	1·20″	2890	4′ 10″	5′ 0″
7″	25″	3″	2·70″	1·55″	3000	5′ 4″	6′ 0″
8″	26″	3″	3·35″	1·90″	3030	6′ 0″	7′ 0″
9″	26″	3″	4·80″	2·75″	4040	8′ 0″	9′ 0″
10″	27″	3″	5·35″	3·10″	4590	9′ 6″	9′ 9″
12″	28″	3″	8·35″	4·80″	4390	11′ 6″	13′ 4″

* These figures include 33 per cent. allowance for general contingencies, and are for working temperatures up to 375° F.

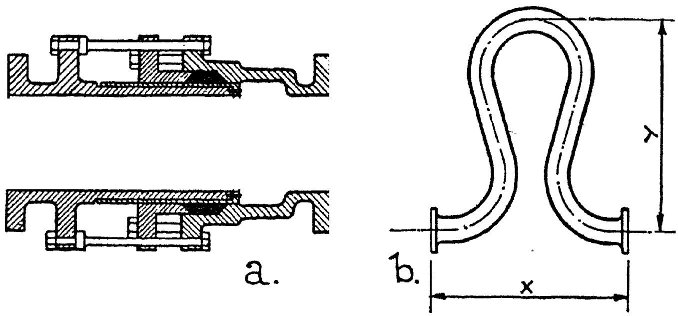


FIG. 139.—Expansion Joints for Heating Pipes.
(*a*) Sliding Type. (*b*) Horseshoe Type (Stewarts & Lloyds).

Various data relating to the types of joint shown in Fig. 139 are given in the Table XLII. See more detailed discussion in Chap. XIV.

Cost of Installations*—Table XLIII gives the total cost of hot-water heating systems in a variety of actual buildings, and the prices are analysed

TABLE XLIII

COST OF HEATING INSTALLATIONS

No.	Type of Building	No. of Occupants	Type of Heating	Method of Firing	Cost of Heating Installation	Gross Building Cube (ft.^{3})	Cost Per Cubic Foot
					£		d.
	Bank Head Office - -		Embedded Panel (Copper)	Oil	22,700	3,129,000	1·7
	Bank - - - - -	200	do.	Oil	6,200	915,000	1·63
	Government Office Building	300	do.	Oil	8,500	1,127,000	8
	Church - - - -	150	do.	Oil	600	108,000	1·33
	Secondary School - -	400	do.	Coke	3,280	523,000	1·5
	do.	400	Embedded Panel (Iron)	Coke	2,580	523,000	1·2
	Hospital (Single Storey Blocks)	240	do.	Coal	10,800	1,048,000	2·47
	Private Office Building -	350	Ray-Rads.	Coke	5,000	962,000	1·25
	Church - - - -	1000	do.	Oil	4,950	933,000	1·28
	College - - - - -	600	Radiators	Oil	3,200	584,000	1·3
	Town Hall - - -	100	do.	Oil	1,580	331,000	1·15
	Hotel - - - - -	80	do.	Coke	1,450	286,000	1·22
	Flats (good class) - -	(600 Flats) 1200	do.	Coke (Automatic)	13,880	4,600,000	0·72
13	Flats (cheap type) -	80	do.	Coke	500	330,000	0·36
14	Factory - -		Unit Heaters	Coal	1,050	350,000	0·72
15	Factory - -		Unit Heaters (H.P.H.W.)	Coal (Auto-stokers)	45,000	35,500,000	0·3
16	Sports Stadium -	11,000	Unit Heaters and Radiators	Coke		4,640,000	0·21
	Residential Club -	350	Radiators	Coke (Automatic)	5,100	978,000	1·25

* See note in Preface as to costs.

into cost per foot cube. These figures may be useful when comparing costs for similar buildings. In each case the type of heating and method of firing the boilers is stated. The costs are inclusive of all piping, radiators, boilers, pumps, insulation, tanks and all fittings complete, but do not include builder's work. The latter usually amounts to about 5 per cent. of the cost of a heating system in a new building, but may be more with an old one.

Note on Pipe-Sizing Formulae—The question of the flow of fluids in pipes has been studied by many experimenters, and a variety of formulae have been evolved, of varying accuracy and refinement. Broadly speaking these fall into two categories:

(*a*) *Approximate* formulae of the form

$$\text{Loss of head} = K \, . \, \frac{V^m}{d^n},$$

where K is taken as a constant,
V is the velocity of flow,
d is the pipe diameter,
m, n are indices determined experimentally.

These formulae are designed to be used for certain restricted ranges of conditions, i.e. water at a given temperature, pipes of specified material and internal condition, also of a certain range of sizes, and so on.

(*b*) '*Rational*' formulae which are applicable to any fluid under any conditions. For circular pipes running full, the form of the equation is

$$\text{Loss of head per unit length} = Cf \frac{W^2}{y \, . \, d^5},$$

where C is a constant,
W is the weight of fluid flowing per unit time,
y is the density of the fluid,
d is pipe diameter,
f is a factor which varies itself according to
the pipe diameter,
the velocity,
the density,
the viscosity of the fluid.

It will be seen that the constants in the Rational formulae are true constants, whereas those used in the approximate formulae are actually average values (for the conditions considered) of factors which themselves vary.

Table XXXIX is based on Rietschels' formula, which is of the approximate type, and is applicable to flow of water at 140–180° F., in steel pipes, diameters ½ in.-12 in.

CHAPTER X

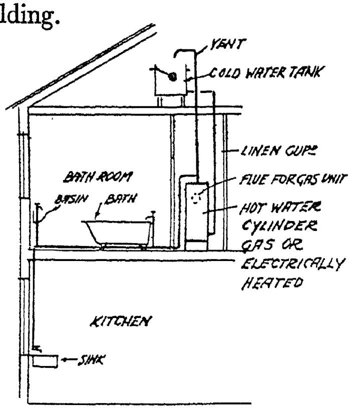
Hot-Water Supply

Local and Central Systems—

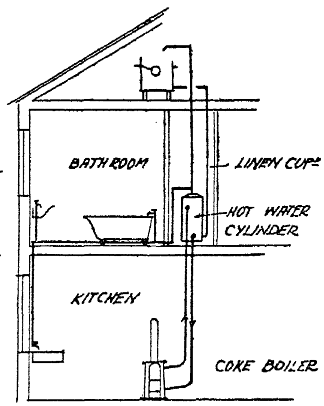
Hot-water supply systems may be local or central exactly as in the case of heating systems.

Local systems are those in which the water is heated immediately adjacent to the bath, basin or sink where it is to be used. A central system is one in which the water for a whole building or group of buildings is heated at one central point and is conducted to the various fittings through a system of pipes.

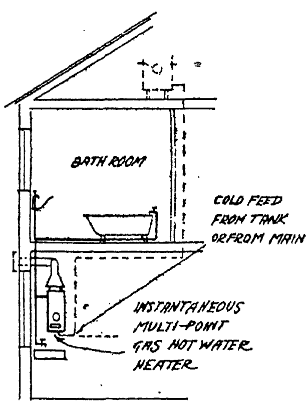
Choice of System—The choice of the right system will depend on the kind of building.



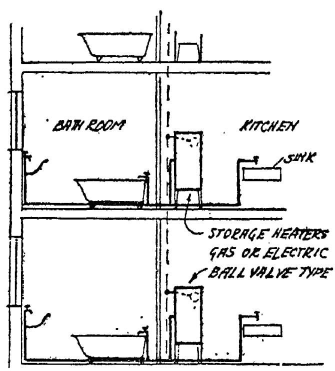
(i) Gas or Electric Storage Heater.



(ii) Coke-fired Domestic Boiler.



(iii) Gas Instantaneous Heater.



(iv) Self-contained Units for Flats, etc.

Fig. 140.—Alternative Systems of Hot-Water Supply to Small Houses or Flats.

Small Houses or Tenements. These will be built in vast numbers under the National Housing Scheme. Fig. 140 shows different methods of dealing with this case. Electricity or gas offers many advantages as compared with solid fuel firing in a boiler. These include:

Cleanliness.
Convenience.
Absence of labour in stoking and ash removal.
Uniformity of temperature of hot water.
Absence of a chimney.

In favour of the small domestic boiler system, fired with coke or anthracite, should be mentioned:

The gentle warmth it gives to kitchen and house.
It provides a means of disposal of a certain amount of refuse.
One or two small radiators may be added for halls, etc., at little extra cost or fuel consumption.

As regards cost of running, the following example will show that there is little difference at the prices of fuel stated.

Consumption of hot water, say,	40 galls. per day
Temperature of hot water	140° F.
„ „ cold water	50° F.
Rise	90° F.

B.T.U.'s *per annum*

Heat to heat water (take 350 days per annum) $= 40 \times 10 \times 90 \times 350 = 12{,}600{,}000$ B.T.U.'s.

Heat lost by radiation:

Electricity, storage cylinder type—
500 B.T.U.'s/hr. × 24 hrs. = 350 days = 4,200,000

Gas, storage type heater—
1000 B.T.U.'s/hr. × 24 hrs. × 350 days = 8,400,000

Coke, boiler and cylinder—
5000 B.T.U.'s/hr. × 24 hrs. × 350 days = 42,000,000

Heat Input and Cost.

Electricity. Total B.T.U.'s/annum = 16,800,000

100% efficiency. Units $= \dfrac{16{,}800{,}000}{3415}$ = 4,900 units

at ½d. per unit = £10 2s. per annum.

Gas. Total B.T.U.'s/annum = 21,000,000
75% efficiency, therms = 210
at 10d. per therm £8·75 per annum

Coke. Total B.T.U.'s/annum = 54,600,000
50% efficiency, coke 12,000 B.T.U.'s per lb.

$$\frac{54{,}600{,}000 \times 100}{12{,}000 \times 2240 \times 50} = 4 \text{ tons}$$

at 50s. per ton = £10

On this basis the relative costs are then:

Electricity, £10·2; *Gas*, £8·75; *Coke*, £10 per annum.

Blocks of Flats. The individual or local system in each flat has the advantage that, as the heat used has to be paid for by the user, a greater economy is exercised.

In the better-class blocks the centralized system is often preferred, as it avoids the space and obstruction caused by separate units. The initial cost of a large central system may be less than a quantity of small units, when cost of cabling or gas piping are included. A boiler plant will probably be provided in any event for central heating, so there is no additional labour for the central hot-water supply system.

Office Blocks, Factories, etc. If the lavatories are few and widely separated, the advantage will be with the small separate local hot-water supply units, electrically or gas heated.

If the lavatories are arranged in compact blocks, the central system will be more economical, and will give a greater reserve for sudden peak load draw off.

Hospitals, Hotels, Institutions. In this class of building the central system is preferable for the following reasons:

Great reserve for heavy demands.
No apparatus to be tampered with.
Maintenance of numerous small units avoided.
The ratio of heat supplied as hot water to radiation losses is greater, hence the use of cheaper fuels shows up to advantage.
The heat is often supplied from a central steam plant providing all heat requirements, and if electric generation is included, use is made of the waste heat.

CENTRAL SYSTEMS

Separate and Combined Systems—If the water for hot-water supply is drawn from the central heating system many disadvantages accrue. The principal one is the furring or corrosion which will take place in pipes, boilers and radiators due to the water being constantly changed. This does not occur when the heating system is kept separate, as the same water is then used over and over again. Further (in the former case) the boiler must be of a type which can be easily cleaned out, and these are usually not the most efficient.

Another disadvantage is that the temperature of the heating water may require to be widely different from that most suitable for hot-water supply, and such difference will not be possible since all the water comes from the same source.

It is therefore most desirable that the systems should be kept separate, which may be accomplished either by the use of a calorifier in a *Combined System* or by providing entirely separate boilers and piping for each system.

Fig. 144 illustrates the former, and Figs. 140 (*a*) to 143 the latter method.

An exception occurs in very small domestic systems in which a small boiler in the kitchen supplies hot water for the taps and also feeds a radiator in the hall and possibly one in a bedroom. Here the apparatus is so small that the disadvantage may not be serious.

The diagrams Figs. 140 (*a*) to 145 illustrate various typical arrangements of hot-water supply systems.

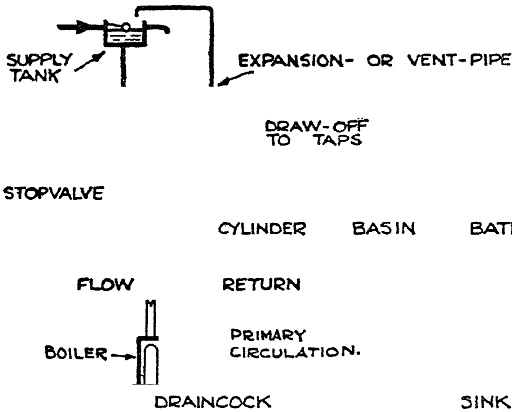


FIG. 140 (*a*).—Simple Direct Hot-Water System.

Separate Systems—Fig. 140 (*a*) shows a simple direct hot-water system in which there is a continuous circulation from the boiler to the cylinder and back to the boiler whereby the water in the cylinder is gradually raised to the required temperature. In this system it is desirable

(*a*) That the cylinder shall be as near the boiler as possible so as to offer the least resistance to this circulation.

(*b*) That the flow pipe from the boiler shall be high up on the cylinder and the return taken from the bottom of the cylinder so as to avoid mixing the hot and the cold water, to give as quickly as possible a layer of hot water at the top of the cylinder, available for drawing off before the whole cylinder has necessarily been heated.

(*c*) That the cold-water supply should be introduced either to the bottom of the cylinder or into the return pipe from the cylinder to the boiler, so that it does not mix with the hot water at the top of the cylinder.

(*d*) That there should not be a hot-water storage-tank at the top of the system additional to the cylinder, as this takes much longer to become hot, and the draw-off will then be partly from the hot cylinder and partly from the tank at the top, which may not be hot, with less satisfactory results.

In the case of the system shown, the pipes from the cylinder to the baths, sinks, etc., give a direct draw-off, not constituting circulating mains. This is suitable for cases where these pipes are short, as in some private houses, but would be quite unsuitable where a long travel is necessary, since in that case a large quantity of cold water has to be drawn off before the hot water is obtained.

Fig. 141 shows the same system with 'secondary' circulating mains with gravity circulation. The system contains all the features of that shown in Fig. 140 (*a*).

With this system, hot water will be obtained almost immediately at each draw-off point. It also enables heated towel rails, circulating coils for linen cupboards, and any other fitment which requires to be kept hot summer and winter, to be connected and to work satisfactorily independent of the draw-off of hot water.

For the system to work satisfactorily, the circulating mains must be correctly sized, having regard to the gravity head due to the cooling surface

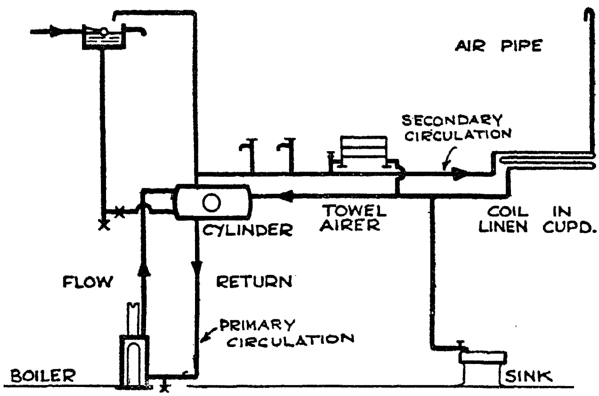


FIG. 141.—Direct Hot-Water System with Secondary Circulation.

of the mains and their attachments, taking into account the frictional resistance of the pipes, exactly as for a heating system.

The return mains should connect into the cylinder not lower than one-third down from the top, so as to prevent a flow of cold water being drawn into the return pipe when a tap is opened.

It must be remembered that in a cylinder arranged in this manner there is generally a very clear line of demarcation between the hot and the cold water, the upper portion of the cylinder normally being filled with hot water and the lower portion with cold, and in a well-designed system these do not mix and do not transfer heat from one to the other except by conduction, which is a very slow process. The effect of the primary circulation from boiler to cylinder is to draw the cold water through the boiler and discharge it into the hot-water layer at the top of the cylinder so that the line of demarcation is gradually lowered. As soon as there is a draw-off of hot water, on the other hand, this takes place from the upper portion of the cylinder only, and the dividing line between the two waters rises, due to the entering cold water pushing the hot water upwards.

Where a cylinder is uninsulated, it is frequently possible to tell by touch where the line of demarcation exists at any instant, within ½ in. or so.

The system is suitable for a large house or even for very large buildings where the height is great in comparison with the horizontal runs, and consequently the gravity head is sufficient to promote healthy circulation.

Fig. 142 shows a similar system with pump circulation. With this system it is still desirable that the circulating main should deliver about

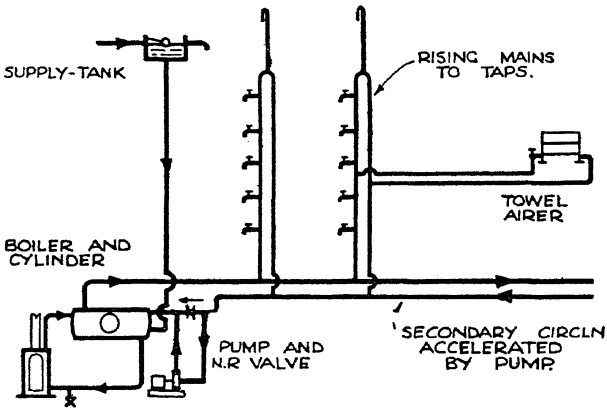


FIG. 142.—Direct Hot-Water System with Pump Circulation.

two-thirds up the height of this cylinder, since, although there is now no chance of cold water being sucked back in the case of a draw-off, owing to the action of the pump, it still remains undesirable to mix the cold water at the bottom of the cylinder with the hot water at the top. The cold water should still be introduced as near the bottom as possible.

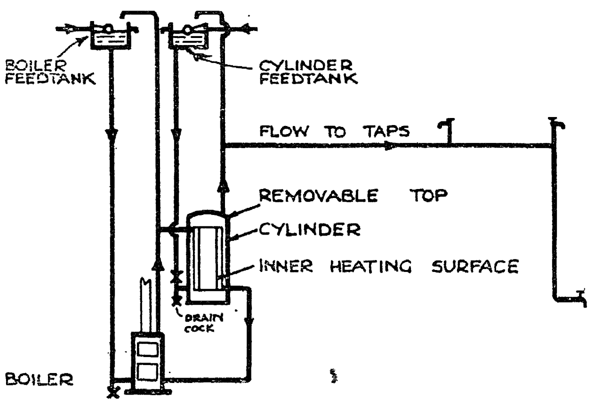


FIG. 143.—Simple Indirect Hot-Water System.

This system is suitable for the largest buildings, or for scattered blocks of buildings operated from a central source.

Fig. 143 shows a simple hot-water system with direct draw-off, but embodying a calorifier.

The boiler circulates hot water to the coils or inner heating surface of the calorifier, which returns it to the boiler in what is known as the primary circulation. This is a closed circuit, and water in this circuit does not communicate in any way with the water which is delivered to the various taps, baths, basins, etc. Cold water envelops the coils or inner heating surface of the calorifier, and is warmed thereby, and the top of this calorifier is then connected to the draw-off mains, this constituting the secondary system.

The advantages of this system as compared with that shown in Fig. 140 are principally that furring of the boiler and the mains immediately connected to it are avoided, even when hard waters have to be used, with the result that several more efficient boilers become suitable, and the efficiency is at all times higher.

It also allows the use of iron boilers and mains for the primary system with corrosive waters, which, if used in the boiler without a calorifier, would cause discoloration and excessive corrosion. The use of special

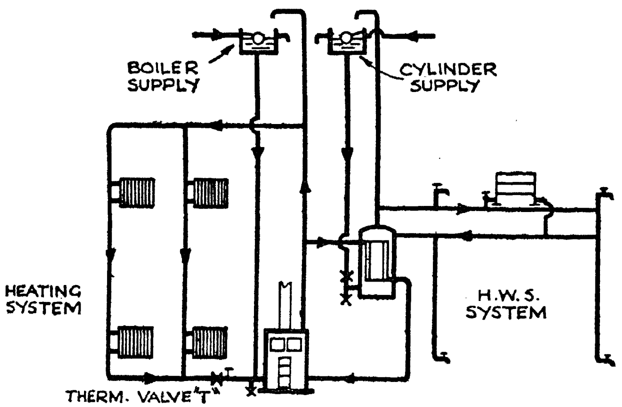


FIG. 144.—Combined Indirect Hot-Water and Radiator System.

metals not likely to be corroded is still a necessity in the secondary system. With hard water, furring may still occur to a limited extent, but will chiefly take place on the outside of the coils in the calorifier, which can be so arranged that it can be cleaned much more easily than can most types of boilers.

As the calorifier usually serves all the purposes of the cylinder, these advantages are often obtainable without much additional cost.

Combined Systems—Fig. 144 illustrates an indirect hot-water supply combined with a radiator system. In this case, the hot-water supply is shown with circulating mains, though it could, of course, as well be with a simple draw-off system, as in Fig. 140 (*a*).

It will be noticed that this system extends the primary circuit to the radiators, and still remains a closed system where the same water is constantly re-circulated, and therefore operates without the disadvantage of

furring with hard water or corrosion with soft water, associated with serving a radiator system directly from the hot-water supply.

The only disadvantage as compared with running the two entirely separately from two boilers, is that in medium weather it may be more difficult to reduce the temperature in the radiators quite so effectively without detrimentally affecting the temperature of the hot-water supply. This can, however, be overcome to a large extent by throttling the circulation in the heating system at the point marked '*T*', which can, if desired, be done under thermostatic control, so as to be self-operating, according to the temperature in any selected room.

This system is quite suitable for small residences where the cost of a completely separated system of heating and hot-water supply is to be avoided.

It is particularly unsuitable where a large boiler has to be provided for an extensive heating system since, in summer, the load of the hot-water service will be much too small for the boiler, which will consequently be difficult to control and keep at a low enough temperature.

Head-tank System—Fig. 145 introduces the old type of hot-water supply system, which is mentioned only with the object of drawing attention to its many defects, because it is still favoured by some plumbers.

In this system, the boiler is at the bottom and the hot-water tank at the top, and when no draw-off occurs, the water gradually rises from the boiler to the head-tank, which then acts in the same way as a cylinder, the level between the hot and the cold water gradually falling in the normal manner.

The circulating pipes between the two are, however, extremely long, so that the circulation is sluggish, and it is frequently necessary for the boiler to get up to boiling point before much circulation occurs.

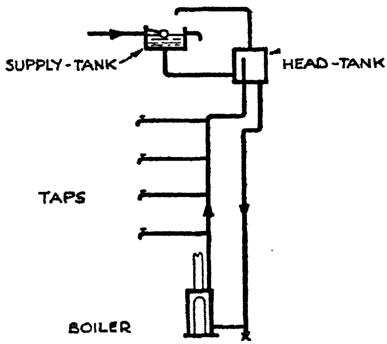


FIG. 145.—Old-type Head-Tank System (not recommended).

When a draw-off occurs, it is largely a matter of chance whether the water will be taken from the boiler or from the top tank, with the consequence that even though the boiler has been alight for a considerable time, it may happen that a mixture of hot and cold water is delivered to any individual tap, since the cold water runs down the return pipe and passes the boiler so quickly as to benefit very little by such passage before reaching the draw-off point.

As there is no reason why this system should be more economical than the one shown in Fig. 140 (*a*), there is no excuse for its continued use.

Capacity of Storage Cylinders—The decision as to the size of cylinder and boiler necessary depends on considerations quite different from those

which apply in a heating system where, as a rule, a fairly constant quantity of heat has to be supplied, and where this quantity is susceptible of fairly accurate calculation, the radiation surface and similar quantities are capable of close estimation.

A hot-water supply system, on the other hand, as a rule functions intermittently. For example, in a normal installation, there is very little hot water required except when hot baths are drawn off in the morning and in the evening, with a certain amount of water taken intermittently by basins, kitchen wash-up, etc.

A calculation of the total quantity of heat required in twenty-four hours will therefore give no criterion of the capacity of cylinder required, unless the cylinder were designed to give a twenty-four-hour storage, which would usually be grossly uneconomical.

In general, the more generous the cylinder capacity, the smaller the boiler power that may be used, as it has a long time in which to catch up the draw-off at peak load. On the other hand, the more sluggish will be the raising of the temperature when starting from cold, or when the cylinder temperature has for any reason been allowed to fall below normal. Between these two extremes a compromise is to be effected.

In many installations it will be found reasonable to give the cylinder a capacity equal to the maximum draw-off of hot water in any one hour at peak-load conditions, and the boiler may generally then be sized on a basis of heating this quantity of water up to the desired temperature in some longer time, depending on the installation. In many cases it will be adequate if this heating takes $1\frac{1}{2}$ to 2 hours, but where there is little draw-off between the peak-load conditions, this period may be further extended, and, on the contrary, where the supply approximates more to a continuous one, it may need to be shortened.

There are special cases where a cheap supply of waste heat is available at a slow and almost continuous rate, as, for example, when the system is combined with a private generating station. In such a case a much larger cylinder capacity may be desirable, so as to get the maximum benefit from the source of heat.

TABLE XLIV

CAPACITY OF VARIOUS STANDARD FITTINGS

	Capacity in Gallons	Temperature usually required, °F.
Lavatory basin, normal filling	1	
full	2	
Sink, normal	5	
„ large	10	
Bath, average	30	100°
Shower Bath, spray type	$1-1\frac{1}{2}$ g.p.m.	90°
„ „ 6″–7″ rose type	7–8 g.p.m.	

TABLE XLV

HOT-WATER CONSUMPTION IN VARIOUS TYPES OF BUILDING

Type of Building	Consumption per Day per Occupant in Galls. (Water at 150° F.)	Peak Consumption per Hour per Occupant in Galls.
School (Boarding)	20	4
Block of Flats	25 to 35	10
Hotel	20 to 30	10
Factory (excluding process work)	4	2
Block of Offices (including cleaning)	5	2
Hospital (Infectious)	50	10
Hospital (Sick)	35	7
Hospital (Mental)	25	5

In most installations hot baths constitute the peak load for the hot-water supply system, especially in hotels and similar buildings. In any case, it is necessary to consider whether during such peak loads a supply is also required for kitchen, basins, etc.

Table XLIV gives the capacities of various standard fittings, and Table XLV the approximate figures of consumption for various types of building, as a guide to the cylinder capacity.

As an example, in a hotel with 100 rooms, each having its own bathroom, each bath might be used twice in one hour, so that it would be desirable to allow

$$100 \times 2 \times 30 = 6000 \text{ galls. at } 100° \text{ F.}$$

The storage cylinder may be assumed to be heated to 150°, so that the storage necessary will be reduced and the volume in the bath made up with cold water. Taking the latter at 50°, the quantity of hot water at 150° to give a mixture at 100° will be half of 6000 galls., i.e. 3000 galls. at 150°.

To this storage of 3000 galls. something should be added for the kitchen (probably in use at the same time), making the total cylinder capacity required perhaps 4000 galls.

Here it should be noted that the 4000 gallons is the hot water at 150° actually required, but as there is bound to be some mixing with the entering cold supply, something should be added to arrive at a satisfactory storage capacity. It is usual to take the effective storage at 70 per cent. of the actual to allow for this incomplete stratification. Thus in the example, actual storage to be provided

$$= 4000 \times \tfrac{100}{70} = 5700 \text{ galls.}$$

Public bathrooms not attached to a particular bedroom may, of course, be used much more frequently in one hour, particularly when the number of bathrooms is small compared with the number of bedrooms. It is impossible to lay down even tentatively any general rule without considering most carefully the particular circumstances of each individual application, and the special circumstances and use for which each installation is intended needs to be made a special study before a proper estimate can be made.

In very large installations it sometimes happens that the capacity of the circulating mains adds materially to the hot-water storage and may be considered as part of the cylinder capacity. This is particularly true when the pipes have been generously sized so as to give an ample circulation without the use of the pump. In the same way, the water contents of the boiler may be considered as part of the cylinder contents. This makes little difference in small installations, but may be an important factor in large ones, particularly if boilers of the Lancashire, Cornish, or other type of large water capacity are used. Indeed, in a large mental institution, recently completed, the capacity of boilers, plus mains, is utilized without a cylinder at all.

Table XLVI gives the capacities in gallons of cylinders of various dimensions, assuming flat ends.

Boiler Power Required—The boiler power is arrived at, as already stated, by assessing the allowable re-heating period. In addition to the heat required for raising the hot water to the required temperature, the losses due to radiation from boiler, mains, cylinder and fittings must be properly allowed for, and in some systems this bears a very high ratio to the total heat required. In the above example, assuming a re-heating period of 2 hours, and radiation losses from the system of 250,000 B.T.U.'s per hour, the boiler power is estimated as follows:

Heat to raise water temperature,

4000 (galls.) $\times$ 10 (lbs./gall.) $\times$ 100 (i.e. $150° - 50°$) = 4,000,000 B.T.U.'s.

Allow 2 hours re-heating period,

$$\frac{4{,}000{,}000}{2} = 2{,}000{,}000 \text{ B.T.U.'s/hour.}$$

$$\text{Radiation} = \quad 250{,}000.$$

$$\text{Net boiler power} = \underline{2{,}250{,}000} \text{ B.T.U.'s/hour.}$$

TYPES OF BOILER FOR HOT-WATER SUPPLY

Boilers which are to be suitable for direct hot-water supply, in addition to all the good points of boilers suitable for heating, must be of such a type as can easily be descaled, i.e. have their interior surfaces accessible for cleaning and the removal of such deposit of scale as may be formed thereon. It has already been explained that in an H.W.S. system where the water is constantly changing, the incidence of such scaling is great in comparison with an ordinary heating boiler, where the same water is recirculated and loses any lime and similar deposits the first time it is used, and thereafter continues to circulate without any further deposit being formed. It is, of course, obvious that the importance of this descaling will depend chiefly on the hardness of the water used, and the point does not arise with soft waters. Such waters are, however, often corrosive in their nature, and it would then be inadvisable to use an iron boiler except for the primary

TABLE XLVI

CAPACITY OF HOT-WATER SUPPLY CYLINDERS IN GALLONS

Diameter of Cylinder	Length of Cylinder																			
	1′ 0″		1′ 6″	2′ 0″	2′ 6″	3′ 0″	3′ 6″	4′ 0″	4′ 6″	5′ 0″	6′ 0″	7′ 0″	8′ 0″	9′ 0″	10′ 0″	11′ 0″	12′ 0″	13′ 0″	14′ 0″	15′ 0″
1′ 0″	4·9	6·1	7·3	9·8	12	15	17	19	22	25	29	34	39							
1′ 3″	7·7	9·6	11·5	15	19	23	27	30	34	38	46	53	61	69						
1′ 6″	11·0	13·8	16·5	22	27	33	38	44	49	55	66	77	88	99	110					
2′ 0″	19·6	24	29	39	49	59	68	78	88	98	118	137	157	176	196	216				
2′ 6″		38	46	61	76	91	107	122	138	153	184	214	245	275	306	336	367			
3′ 0″			66	88	110	132	152	176	198	220	264	308	352	396	440	484	528	572		
3′ 6″				120	150	180	210	240	270	300	360	420	480	540	600	660	720	780	840	
4′ 0″					196	235	275	314	353	392	470	549	628	708	785	862	945	1020	1100	1180
4′ 6″						298	347	398	447	496	596	695	794	894	991	1090	1190	1290	1390	1490
5′ 0″							428	490	550	612	734	857	979	1100	1220	1350	1479	1590	1710	1830
5′ 6″								592	667	741	889	1040	1190	1330	1480	1630	1780	1930	2080	2230
6′ 0″									794	882	1060	1230	1410	1590	1765	1940	2120	2280	2470	2650
7′ 0″										1200	1440	1680	1930	2160	2400	2650	2880	3130	3370	3610

circulation in conjunction with a calorifier, in which case any boiler suitable for heating becomes equally suitable for this purpose.

In the case of boilers which are to be used with a water of ordinary hardness (say 8° of hardness or upwards) without a calorifier (i.e. connected directly to the cylinder and draw-off mains), the following types may be considered suitable.

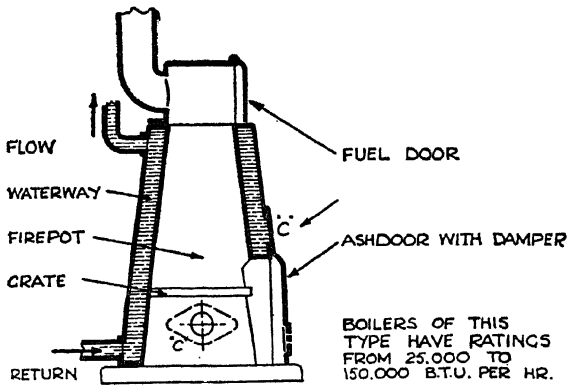


FIG. 146.—Typical Cast-Iron Domestic Boiler.

(*a*) **Cast-Iron Domestic Boilers**—(Fig. 146)—Boilers of this type are generally provided with two or three cleaning covers or mud holes at the top and the bottom, from which descaling can be carried out with special tools. It must, however, be admitted that there are portions of the surface

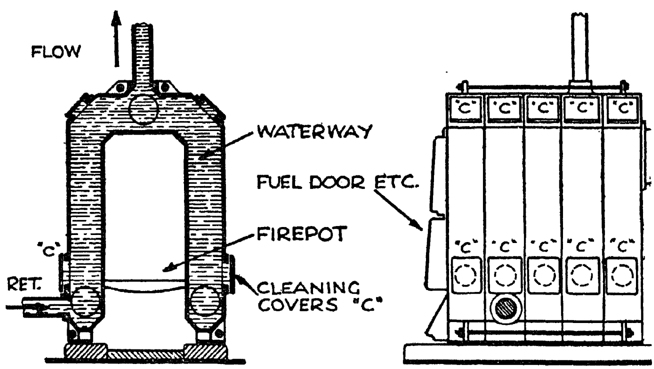


FIG. 147.—Typical Cast-Iron Sectional Hot-Water Supply Boiler. (Ratings up to 400,000 B.T.U./Hr.)

which remain difficult of access, and with very hard waters the boiler may need to be replaced after service of ten to twenty years, depending on the hardness of water. The forcing of the boiler makes for greater formation of scale, and it is therefore better to have a boiler of ample size

and use it without the need of forcing. This also leads to less clinkering and greater convenience in running generally.

Where dealing with corrosive waters cast-iron boilers can be treated by the Bower-Barff process, whereby their resistance to corrosion is increased with certain limitations. This process consists of heating the iron to a high temperature in the presence of oxygen, producing a skin of magnetic oxide of iron, which is highly corrosion-resisting.

(*b*) **Sectional Cast-Iron Boiler**—(Fig. 147)—These are only to be recommended for direct hot-water supply with hard water when they are of the special type which permits complete descaling. For this purpose they are made with large waterways and large cleaning covers top and bottom on both sides as indicated in the illustration.

All cast-iron hot-water supply boilers suffer from the disadvantage of liability to crack due to the rapid and continuous changes of temperature to which they are subjected, and to the presence of scale producing local over-heating, which is particularly detrimental to cast iron.

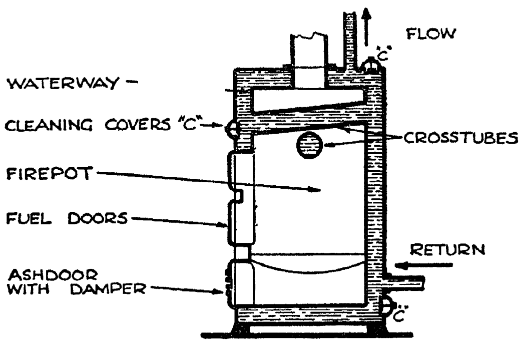


Fig. 148.—Typical Mild-Steel Domestic Boiler.

(*c*) **Mild-Steel Domestic Boiler**—(Fig. 148)—Wrought-iron or mild-steel boilers are suitable for higher pressures than cast-iron boilers, but are generally somewhat more expensive. (The approximate relative costs are shown in Table XLIX, p. 242.) Their great merit is freedom from cracking, and long life. In addition, whereas a cast-iron boiler should not, in general, be used for a head of more than 100 ft., a mild-steel boiler can be designed for any head. In corrosive waters, it is sometimes found that cast-iron boilers are not so susceptible to corrosion as mild-steel boilers, but it depends on the properties of the water. As, however, some types of steel boiler can be treated by the Bower-Barff process, this disadvantage can largely be removed.

Boilers of the mild-steel type can be combined with a cylinder as a part of the boiler, giving a very compact arrangement.

(*d*) **Mild-Steel Sectional Boiler**—(Fig. 149)—These boilers are made in a great variety of types, and have the advantages over cast iron already

mentioned under (*c*) and need to have the special facilities for descaling already mentioned under paragraph (*b*). They are more efficient than the plain one-piece steel boilers owing to the better arrangement of the heating surface.

(*e*) **Copper Boilers**—In soft water districts boilers made of copper are frequently found. Sulphur in the fuel eats away the firepot, which is made thicker in consequence. The cost of an indirect system using a cast-iron boiler is usually cheaper.

(*f*) **Larger Boilers**—The cast-iron sectional domestic boiler is generally not suitable for a rating much above 400,000 B.T.U.'s per hour, and the

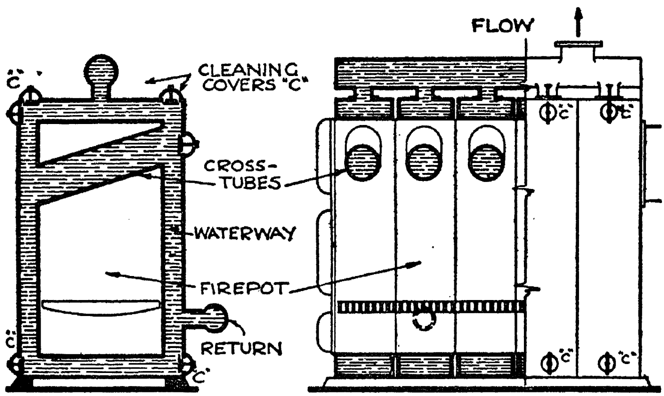


FIG. 149.—Typical Sectional Mild-Steel Boiler. (Ratings up to 1,000,000 B.T.U./Hr.)

mild-steel domestic or mild-steel sectional beyond about 1,000,000 B.T.U.'s per hour. Above these sizes, boilers of a different type are generally to be preferred. Larger boilers for direct use may be of the Lancashire, Cornish, Economic or other easily descaled types, but a calorifier system is usually to be preferred for large duties. It may be served by steam boilers, or by water boilers similar to those used for heating systems as described in Chap. V. Similar safety-valves, mountings, lagging, etc., to those previously discussed would be used with them.

Rating of Heating Surface of Hot-Water Supply Boilers—The rating of boilers for hot-water supply is different from the rating for heating, because in general the water in the boiler is colder when used for hot-water supply than when it is used for heating, and consequently the transmission of heat through the walls of the boiler is greater for each square foot of heating surface, also because it is recognized that the peak load on such boilers is rarely required, so that a less conservative rating is allowable.

At the same time makers' ratings are on the basis of maximum output under test conditions, and continuous working at such rates would lead to unduly high flue gas temperatures and inefficient working, and in any case to rapid furring and consequent burning of the plates or metal of the boiler.

Makers' ratings are generally somewhat as follows:

Domestic Boilers		B.T.U.'s per Hour per Sq. Ft. of Heating Surface
(a) Cast-iron (as Fig. 146)	-	11,000
(b) Cast-iron sectional (as Fig. 147)	-	8,000 to 11,000
(c) Mild-steel (as Fig. 148)	-	7,000
(d) Mild-steel sectional (as Fig. 149)	-	6,000

It will be seen that whilst the cast-iron firepot type boiler (*a*) is rated at 11,000 B.T.U.'s per sq. foot of surface, the mild-steel type is rated at only 7000 B.T.U.'s. The two are similar in having almost all the heating surface primary, i.e. in direct view of the fire, and there appears to be no reason for the difference.

Similar remarks apply in respect of the sectional types, though where these boilers have secondary or tertiary heating surface a lower overall rating is taken by the makers.

The point to be noted is that where the hot-water demand is more or less continuous, the mild-steel boiler ratings may be worked to, but this is not the case with cast-iron boilers, which should be rated down as for mild steel. Only if the maximum output is infrequently required should the higher figures for cast-iron boilers be worked to.

Apart from these considerations, it is desirable to allow a margin of 10 per cent. or so over the estimated boiler power to cover possible contingencies such as under-estimation of the draw-off. Thus, in the example on p. 232 a suitable boiler power would be

$$2{,}250{,}000 \times 1{\cdot}1 = \text{say, } \underline{2{,}500{,}000} \text{ B.T.U.'s per hour.}$$

Rating of Grate Area of Hot-Water Supply Boilers—Considering the rating in terms of grate area, a good output from a small grate necessarily involves a high rate of combustion leading to clinkering. Most catalogue ratings appear to be based on combustion-rates of from 9 to 11 lbs. of solid fuel per sq. foot of grate area. If clinkering is to be avoided, and the boiler is to run for reasonable periods without attention, this rate should not exceed 7 lbs. for continuous working. This is equivalent to approximately 60,000 B.T.U.'s per sq. foot of grate area.

HOT-WATER SUPPLY CALORIFIERS

The advantages to be obtained by the use of calorifiers have already been referred to, and relate principally to the greater ease of descaling and to the wider choice of more efficient boilers, including the magazine types described in Chap. V.

Calorifiers for hot-water supply differ from those for heating in having a much larger shell so as to provide storage. The capacity of the storage is determined as in the case of cylinders already discussed.

The internal heating surface of the calorifier may consist of tubes (see

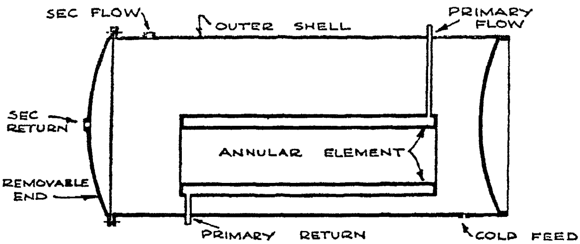
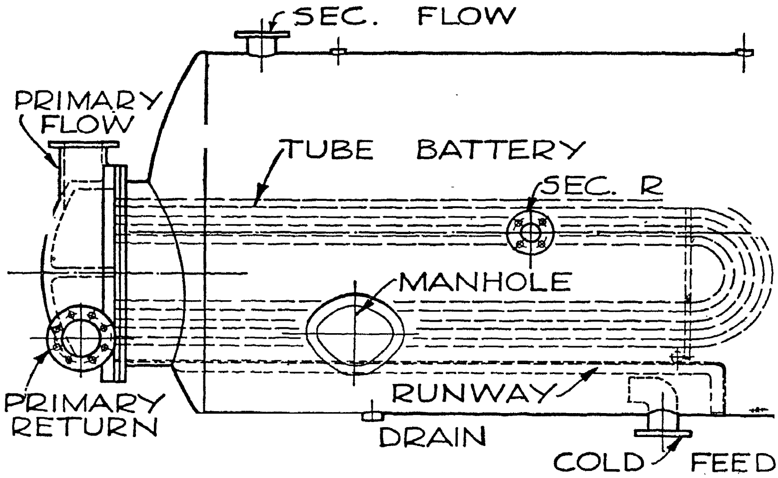
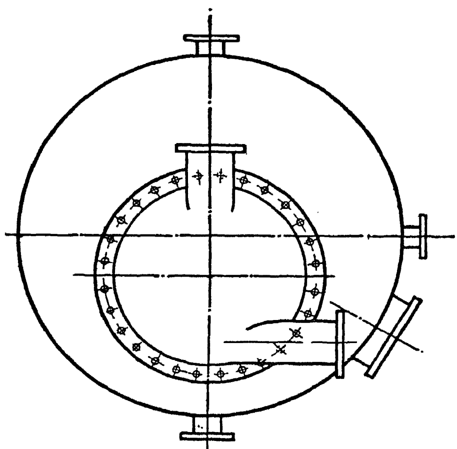


FIG. 150.—Hot-Water Supply Calorifier with Annular Heating Element.



LONGITUDINAL SECTION.



END ELEVATION.

FIG. 151.—Calorifier for Large Water-Water System.

Fig. 151), of radiator sections, or simply of an annular element as shown in Fig. 150. This latter arrangement is preferable for ease of descaling the outside of the element, though a larger heating surface in a given space is possible with tubes or radiators due to their convolutions. With any type of heat exchanging surface the element is best kept near the bottom of the shell so as to promote convection over as great a volume as possible.

Where steam is available, either direct from boilers or as exhaust from engines, it may be utilized in the heating element, and due to its higher temperature will give a much larger transmission per sq. foot, and consequent economy of space, than is possible with water-heated primary.

A typical arrangement for a steam-heated calorifier is shown in Fig. 152.

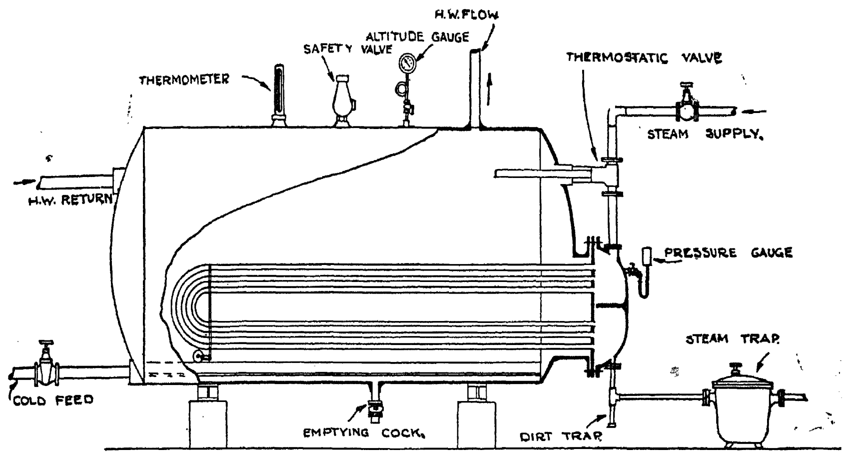


FIG. 152.—Hot-Water Supply Calorifier for Steam-Water System.

Rating of Calorifiers—The heating surface required depends on the temperature difference between the primary water or steam and the secondary water, and on the transmission rate per degree difference.

Water to Water Calorifiers—The point arises as to what is the true temperature difference.

FIG. 153.

Considering Fig. 153, the primary may be assumed to be kept at constant temperature, say 180° flow, 160° return, mean 170°. In the case of the secondary, the temperature in the cylinder at starting from cold is 50°, giving a difference of $170 - 50 = 120°$.

When completely warmed to 150° the difference is only $170 - 150 = 20°$. The mean between these two conditions is when the secondary is at 100° and the difference is then $170° - 100° = 70°$.

If the heating surface is designed on this temperature difference of 70°, it can be shown that the transmission will be in excess of the mean rate for the first half of the temperature rise, but below the mean rate for the second half where the temperature difference and transmission rate is becoming continually reduced. The excess over the first half is not sufficient to balance the deficit on the second half.

The difference between the arithmetic mean temperatures of primary and secondary water may still be used if the transmission coefficients are suitably adjusted.

This is the basis of the coefficients given in Table XLVII below. A primary temperature of 160–180° F. is assumed, and a secondary of about 100° F. mean, with free convection.

For pipe coil surfaces the velocity of flow in the tubes should be calculated to arrive at the appropriate coefficient. In the case of annular or radiator surfaces the velocity is indeterminate and the figures given refer to normal gravity circulation.

Steam-Water Calorifier—Where steam is used as the heating medium, the temperature of the steam will depend on its pressure (see Chapter XIV on 'Steam Heating', p. 307). The temperature of the water may be taken at the arithmetic mean of the secondary. The difference between these temperatures may then be used with one of the coefficients below.

TABLE XLVII

HOT-WATER SUPPLY CALORIFIERS

Type of Surface	B.T.U.'s per hour per sq. ft. per degree F. difference	
Water to Water:		
Annular	25	
Cast-Iron Radiators	20	
Pipe Coil Iron or Copper	Vel.: 1½″/sec.—29. 6″/sec.—50. 12″/sec.—57	
Steam to Water:	Steam at 250° F.	300°
Annular	120	—
Cast-Iron Radiators	80	—
Pipe Coil, Iron	102	114
„ „ Copper	112	124

Note. These coefficients include an allowance for furring.

Material of Calorifiers—Since the inside of the shell and the outside of the heating element are both exposed to the constantly changing secondary water, it is necessary to provide against the possible harmful effects of the latter. To provide for scaling, the internal element should be removable, or at any rate accessible by means of a removable end; and where the supply is corrosive (as is the case of many soft waters) the calorifier should be made of copper.

Where steel calorifiers may be used it is usual to have them galvanized, or in the case of large vessels, simply painted internally with protective paint, which is renewed at the annual overhaul.

FEED TANKS

Feed tanks are required on hot-water supply systems, to supply the water drawn off from the taps, and in addition to allow for expansion as in the case of a heating system.

The sizing of the feed tank depends to some extent on the supply of water available. If the pressure and flow are good a one-hour storage under peak load conditions will probably suffice, but if poor, sufficient for two or three hours would be advisable.

The feed pipe from the tank requires to be connected to the cylinder so as to prevent mixing with hot water, thus preserving the stratification between the two. This may be achieved by finishing with a tee or bend inside the cylinder when entering through the bottom of the shell (see Fig. 151), or with a sparge pipe perforated at the bottom when entering through one end.

Indirect systems require a feed tank to both boiler and calorifier circuits, and these have been shown in the diagrams already given. That fitted to the primary need only be small, being in fact only an expansion tank, since no water is taken from this circuit beyond evaporation from the tank (which is cold) and any which may be lost when a repair is effected.

COST OF INSTALLATIONS

TABLE XLVIII

COST OF HOT-WATER SUPPLY INSTALLATIONS

No.	Type of Building	Cost of H.W.S. Installation	Building Cube	No. of Occupants	Cost per Cub. Ft.	Cost per Occupant	Remarks
	Tavern	£187 15s. od.	165,000	30	·273d.	£6·26	Galvanized pipe, etc.
	Hotel	£610 16s. od.	286,000	80	·510d.	£7·63	
	Flats (cheap type)	£549 os. od.	330,000	80	·400d.	£6·86	
	Town Hall	£310 3s. od.	331,000	100	·218d.	£3	.. ..
	Secondary School	£329 os. od.	523,300	400		£0·82	Copper pipe, etc.
		£232 10s. od.		400		£0·58	Galvanized pipe, etc.
	College (part residential)	£958 15s. od.	688,000	150	·332d.	£6·35	
8	Flats (good class)	£1798 os. od.	750,000	220		£8·18	.. ..
9	Bank Head Office	£975 os. od.	915,000	200		£4·89	Copper pipe, etc.
10	Private Office Building	£1176 os. od.	962,000	350		£3·36	
11	Government Office Building	£656 1s. od.	1,127,000	300		£2·19	
12	Bank	£1892 10s. od.	2,000,000	350	·227d.	£5·40	
13	Bank Head Office	£2573 os. od.	3,129,000	1100	[illegible]	£2·34	.. ..
14	Private House	£35 os. od.	16,000	4		£8·75	Galvanized pipe, etc.

(See note in Preface as to costs.)

Table XLVIII sets out the approximate cost of the hot-water supply system to various completed buildings, and expressed in terms of the cubic capacity of the building and the number of occupants. The cubic capacity

is measured gross cube, the number of occupants is on the basis of the day-time occupation, and the cost of the supply includes boiler, calorifier, or cylinder and mains taken as far as the taps, but not including taps, basins, baths, or similar fittings.

Table XLIX gives the cost of boilers and cylinders of various ratings, capacities and descriptions, and of piping suitable for hot-water services.

TABLE XLIX

COST OF HOT-WATER SUPPLY APPARATUS

BOILERS	APPROX. COSTS, INCLUDING FIXING, INSULATION & MOUNTINGS, 1934.		
RATING† B.Th.U's/HOUR	CAST IRON DIRECT	WROT IRON DIRECT	C.I. SECTIONL INDIRECT
100,000	£ 20	£ 70	£ 30
200,000	£ 40	£ 90	£ 40
300,000	£ 60	£ 110	£ 60
400,000	£ 75	£ 140	£ 75
500,000	–	£ 170	£ 90
750,000	–	£ 220	£ 130
1,000,000	–	£ 270	£ 170

CYLINDERS & CALORIFIERS	APPROX. COSTS, INCLUDING FIXING, INSULATION & MOUNTINGS, 1934				
CAPACITY GALLONS	SIZE L. x DIAM.	GALVD IRON CYLINDERS	GALVD IRON CALORIFIERS	COPPER CALORIFIERS	WORKING HEAD - FT.
50	42" x 21"	£ 10	£ 20	£ 25	40
100	64" x 24"	£ 15	£ 30	£ 50	40
200	60" x 36"	£ 25	£ 40	£ 90	60
300	84" x 36"	£ 50	£ 70	£ 125	100
400	88" x 42"	£ 70	£ 100	£ 175	100
500	84" x 48"	£ 85	£ 120	£ 200	100
750	120" x 48"	£ 100	£ 140	£ 250	100
1000	99" x 60"	£ 120	£ 180	£ 350	100

PIPING	COST PER FOOT RUN, FIXED, WITH AVERAGE NUMBER OF FITTINGS, UNLAGGED	
INSIDE DIAMETER	GALVANIZED IRON	LIGHT GAUGE COPPER & COMPRESSION JOINTS
½"	1/-	1/1
¾"	1/1	1/4
1"	1/3	2/-
1¼"	1/8	2/9
1½"	2/-	3/3
2"	2/6	5/-
2½"	3/4	6/-
3"	4/-	8/6
4"	5/6	12/6

†All boiler costs based on catalogue ratings.
(See note in Preface as to costs.)

CHAPTER XI

Piping for Hot-Water Supply Systems

Pipe runs for hot-water supply systems fall into three categories, which call for separate consideration. There are:

(*a*) Primary pipes, which have to circulate water steadily from the boiler to the cylinder or calorifier.

(*b*) Secondary pipes and feed pipe which are required to pass water spasmodically and in relatively large quantities when a tap is opened.

(*c*) Secondary circulating pipes, which in addition to (*b*), should circulate enough to make good their own heat-loss, and so keep hot water constantly available at the draw-off points.

Primary Circulation (Direct System)—The sizing of the pipes connecting boiler and cylinder on the direct system should be ample, in order, first, to allow for the furring which will inevitably take place in them if the water is hard, and, secondly, so that the frictional resistance may be so low that a brisk circulation takes place.

The sizes may be calculated in exactly the same manner as already described for a heating system operating by gravity circulation, taking the circulating head as the difference of pressure between the hot rising and cold falling columns from centre of boiler to centre of cylinder, with a temperature drop of, say, 50°. This head, divided by the length of travel of flow and return, including bends, etc., will give the circulating pressure per foot of pipe, from which the pipe size necessary may be determined from Table XXXIX. Alternatively, the sizes based on return 50°, flow 100°, height from centre of boiler to centre of cylinder 4 ft., travel 20 ft. plus 4 bends, may be taken from the following table:

TABLE L

SIZE OF PRIMARY FLOW AND RETURN MAINS FOR 'DIRECT' SYSTEM

Size	Maximum B.T.U.'s per Hour Transmitted
	37,
	56,600
	110,000
	180,000
	260,000
	505,000
	840,000
	1,140,000

A size of 1¼ in. should be considered a minimum even for the smallest boilers owing to the necessity of allowance for furring.

F.H.

It must be remembered that the maximum output of a hot-water supply boiler is required at periods of heavy draw-off—that is, when the bottom of the cylinder is certain to be cold.

As the heating of the water in the cylinder proceeds and the return to the boiler rises in temperature, the flow temperature will also rise, though not by the same amount since the circulating head at higher temperatures will be increased and more water will be passed. Thus, when the water is returning to the boiler at 140° F., an equal heat transmission will be taking place with the same pipe sizes with a flow at 174° F.—that is to say, a 34° rise as against 50° F. At this point, however, the output from the boiler should be reduced so as to meet the radiation losses only, otherwise the temperature will continue to mount, which, in most cases, would be unnecessary and wasteful.

This adjustment of the boiler output is to a small extent automatic, since the efficiency at the higher temperatures will be less than at the lower. Similarly, as the temperature of the system as a whole rises, the radiation losses will increase, until with a system having long mains and a small boiler the output will be completely balanced by the emission, and no further temperature rise will be possible.

It will be seen from the above that the primary circulation is a constantly varying one, and as the maximum duty is required at the poorest circulating temperatures, great care should be exercised in seeing that the piping is kept as short as possible with easy radius bends to avoid undue resistance.

Primary Circulation (Indirect System)—With the indirect system, no allowance need be made in the sizing of the primary circulation for furring since the same water is constantly re-used.

A rapid circulation is essential, as the transmission of heat from the primary to the secondary water through the walls of the heating coils depends on a difference of temperature being maintained right through to the outlet.

The flow and return temperatures may be taken at higher figures than with the direct system, since with a combined heating and hot-water supply apparatus the primary water circulation will be the same as that supplied to the radiators at, say, 180° flow and 140° return in cold weather, or 160° flow and 120° return in milder weather. For the purpose of calculating the hot-water supply primary circulation, the latter temperatures should be assumed in a combined system.

Where the indirect cylinder or calorifier is served from a boiler whose sole duty is to provide hot-water supply, it is of advantage to run this at as high a temperature as possible so as to make the heat-exchanging surface of the coils as effective as possible.

The following temperatures may be assumed with buildings of various heights:

Indirect System Height of Feed and Expansion Tank above Boiler	Primary Circulation Temperature Flow, Deg. F.	Temperature Return, Deg. F.
Up to 50 ft. - - -	180	140
50 to 100 ft. - - -	220	180
Over 100 ft. - - -	240	200

Based on a flow and return temperature of 180° F. and 140° F. respectively, a height from centre of boiler to centre of calorifier coils of 4 ft., and a travel of 20 ft., plus 4 bends, plus equivalent of 10 ft. of pipe for resistance of indirect coil, the heat transmitted by the various pipe sizes is given in Table LI.

TABLE LI

Size of Primary Flow and Return Mains for 'Indirect' System
Flow 180°, Return 140°

Size	Maximum B.T.U.'s per Hour Transmitted
	24,000
	41,000
	61,000
2	120,000
2½	207,000
3	326,000
4	584,000
	960,000
	1,332,000

SECONDARY SUPPLY

Outflow from Taps—The sizes of the pipes from the tank to the cylinder and from the cylinder to the taps depend on the outflow from the latter, and not in any way on the boiler load.

Information is required as to the number and positions of taps and their size, on which depends the volume delivered by each.

From the curves of Fig. 154 may be taken the maximum outflow from taps of the commonly used sizes for various heads from tank to tap level. These curves are the results of tests on an installation in which the taps were connected by short pipes to a relatively large riser at each floor, so that the discharge is greater than would be obtainable in the average case, but the shape of the curves is typical.

Further it is usual and necessary to throttle down the taps at the lower floors of a high building to prevent splashing.

A practical and economical limit for discharge may be taken at 1–2 galls. per minute for a ½-in. tap, and 5 galls. per minute for a ¾-in. tap.

The next stage is to determine the number of taps which will be open simultaneously. Obviously all the taps in a building would not be discharging water at the same instant. For example, it may take a ½-in. tap thirty seconds to deliver enough water mixed with cold for a hot wash, but it will be found that at least a further sixty seconds elapse before the washing is completed and probably a further thirty seconds before the next person turns on the same tap. This is a ratio of 30:120 or 1:4.

Similarly it may take three minutes to fill a bath but at least twenty minutes elapse before the next bather is opening the tap again. This gives a proportion of 3:20, or 1:7.

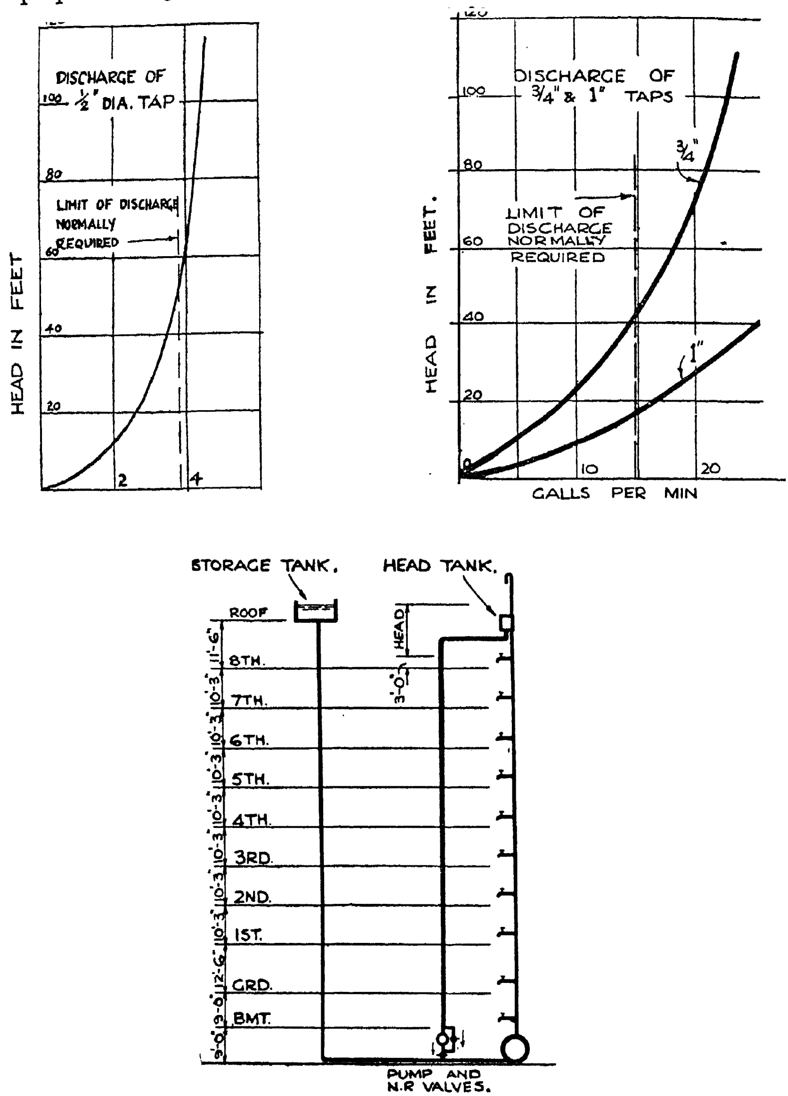


FIG. 154.—Flow from Taps on Riser (see Text).

The proportion of taps open simultaneously must depend to a large extent on the number of taps installed, as in the case of two taps served by a single pipe, both might easily be open at once. On the other hand, if fifty

taps are connected to the same main, the number in use at any moment could never conceivably be fifty.

Fig. 155* shows the percentage of fittings which may be assumed to be in use at any instant, plotted against the total number of fittings connected.

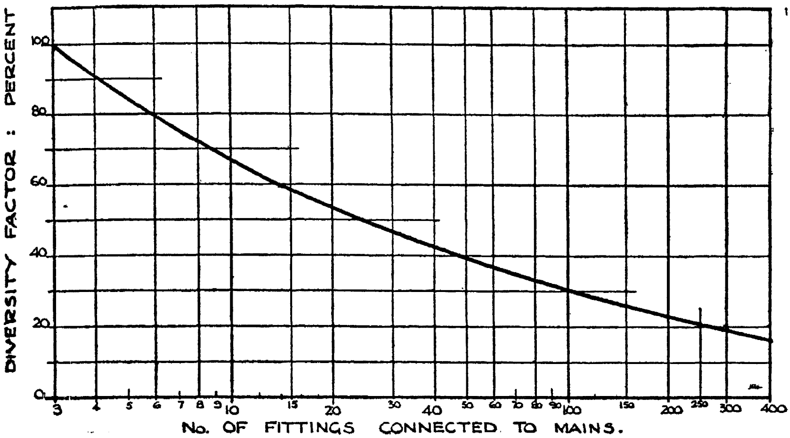


FIG. 155.—Diversity Factors for Hot-Water Supply.

The percentage obtained from this curve should be applied to the total draw-off for all fittings in the building in use together.

Thus, in a building having

200 lavatory basins	($\frac{1}{2}$-in. taps)
10 sinks	($\frac{3}{4}$-in. taps)
40 baths	($\frac{3}{4}$-in. taps)
250 fittings.	

the percentage from Fig. 155 would be 20 per cent.

Total draw-off from

200 $\frac{1}{2}$-in. taps at 1 g.p.m. = 200 g.p.m.
50 $\frac{3}{4}$-in. taps at 5 g.p.m. = 250 g.p.m.

450 g.p.m.

∴ Probable simultaneous draw-off
= 20 per cent. of 450 = 90 g.p.m.

This basis has been found to be quite safe in practice, and its correct application may save considerably in the size of mains called for by the more rule of thumb methods often employed. Obviously the ratio of taps

* It will be noted that the curve is on a logarithmic base. It has been found to give satisfactory results over a large number of installations. Some engineers take the square root of the number of fittings to give the number in simultaneous use. Whilst this may serve in some cases, it is felt by the authors to give too small a result for large numbers of taps.

open simultaneously, to taps connected, will vary for different types of building, and caution is required before applying the method to special cases which must be considered on their merits.

TABLE LII

FLOW IN PIPES FOR HOT-WATER SUPPLY SYSTEMS

a). "F" IS THE HEAD-LOSS IN FT. W.G. PER 100 FT. RUN OF PIPE FOR GIVEN QUANTITY (IN GALLONS PER MIN.) FLOWING IN VARIOUS SIZED PIPES:-

	HEAD LOSS "F"	DIAMETER OF PIPE ½"	¾"	1"	1¼"	1½"	2"	2½"	3"	4"	5"	6"
V = 1 FS	0·5	0·31	1·0	2·1	3·7	6·0	13·0	23	39	83	145	235
	1·0	0·47	1·4	3·0	5·4	8·7	19·0	34	57	122	210	345
	1·5	0·59	1·7	3·7	6·8	11·0	23·5	43	70	153	260	430
	2·0	0·68	2·0	4·3	7·8	12·8	27·5	50	82	175	305	500
	2·5	0·76	2·2	4·9	8·8	14·5	31	57	93	200	345	565
	3·0	0·83	2·4	5·3	9·7	15·8	34	64	102	220	380	620
	4·0	1·00	2·9	6·2	11·4	18·5	40	73	119	260	440	720
V = 3 FS	5·0	1·10	3·3	7·0	12·8	·21	45	82	134	290	500	800
	7·5	1·35	4·0	8·7	15·9	26	56	103	167	365	620	960
	10·	1·6	4·7	10·3	18·8	30	66	121	197	430	700	1140
	15·	1·9	5·7	12·8	23	37	81	133	230	500	830	1310
	20·	2·3	6·7	14·9	27	43	93	163	265	565	940	1480
V = 6 FS	25·	2·6	7·5	16·8	31	47	103	179	295	610	1040	1600
	30·	3·0	8·3	18·0	34	51	112	190	320	660	1130	1730
	40·	3·5	9·5	20	39	59	126	213	375	740	1290	1910
V = 10 FS	50·	4·1	10·7	23	43	65	140	234	420	815	1440	2090
	60·	4·6	11·7	25	47	71	153	254	465	880	1570	2250
	70·	5·0	12·6	28	50	76	165	273	505	960	1690	2400
	80	5·4	13·5	30	54	81	175	288	540	1020	1800	—
	90·	5·7	14·3	32	57	86	185	305	575	1090	1900	—
	100·	6·0	15·0	34	60	90	195	320	605	1150	2000	—

THICK ZIGZAG LINES ARE FOR VELOCITIES OF 1 3 6 10. 20 AND 30 F.SEC WATER AT 165°F.

V = 20 F.S. → V = 30 F.S. →

b) RESISTANCE OF FITTINGS:
TABLE SHOWS EQUIV. FOOT RUN OF STRAIGHT PIPE FOR R = 1, FOR VARIOUS VELOCITIES:—

VEL. FEET /SEC.	DIAMETER OF PIPE ½"	¾"	1"	1¼"	1½"	2"	2½"	3"	4"	5"	6"
1·0	1·4	2·0	3	4	5	7·5	10	13	18·5	22	28
3·0	1·6	2·5	3·5	4·5	6	9	12	15	21	27	33
6·0	1·8	2·5	4	5·5	7	9·5	13	16·5	23	30	36
10·	2·0	3·0	4·5	6	7	10	14	18	25	31	38
20	—	—	—	—	7·5	11	15	20	27	33	40
30·	—	—	—	—	—	—	—	22	29	35	42

GATE-VALVE OR PLUG-COCK	R = 0·5	TEE, STRAIGHTWAY	R = 1·0
BEND, SHORT RAD.	R = 2·0	TEE BRANCH	R = 1·5
BEND, LONG RAD.	R = 1·0	ENLARGEMENT	R = 1·0

The next stage in the sizing of the pipes is to determine the head available for delivering the water. This is the gravity head or distance between the lowest water level in the supply tank and the topmost tap. Obviously the topmost must be taken as this is bound to be the worst case with the upfeed system.

This head must deliver the calculated quantity of water through the pipes, and as the length of the latter can be measured from the plans, making due allowance for bends, etc., as for a heating system, the available head per foot run of pipe may be calculated. Knowing this figure, the sizes of pipes necessary for the estimated deliveries can be read from Table XXXIX, though it must be remembered that the latter is expressed in lbs. per hour for pressure losses expressed in inches of water column, whereas the calculations above described have been on the basis of gallons per minute and feet head. For convenience Table LII gives this data over the range normally required, expressed in gallons delivered and resistance in feet of water column per 100 foot run.

The following example will help to make the above methods clear.

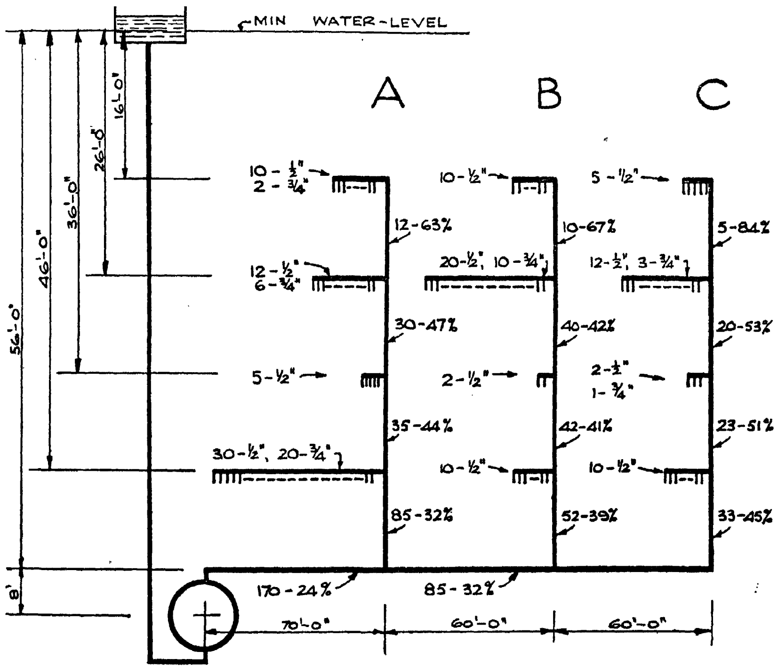


FIG. 156.—Up-feed System (Returns not shown).

Sizing Secondary Flows—Fig. 156 represents an up-feed system supplying ½-in. and ¾-in. taps as noted. The secondary returns are omitted for clarity.

The total number of fittings for each section of the riser and for the main may be found by progressive addition, as shown. The diversity factors are then taken from Fig. 155 for the appropriate number of fittings, and noted against each section of the piping.

The next step is to find the maximum possible draw-off at each point.

The progressive totals are therefore taken out again, but keeping the various sized fittings separate, and the possible draw-off marked on (at 1 g.p.m. for ½-in. taps and 5 g.p.m. for ¾-in. taps) as Fig. 157.

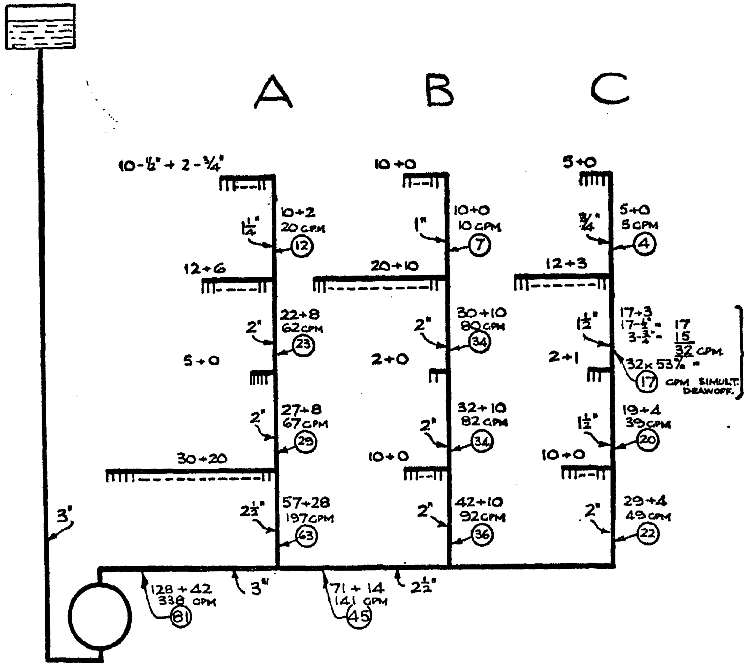


FIG. 157.—Up-feed System (Returns not shown).

The diversity factors are then applied to the draw-off figures, and the probable simultaneous draw-off arrived at. These figures are shown in circles on Fig. 157.

For the sizing of the pipes we require to know the head H and travel T for the worst case, i.e. for riser C. This is for the top floor taps where:

$$T = \text{(say) } 70 \text{ (drop)} + 50 \text{ (rise)} + 190 \text{ (horizontal)} + R = 310 + R.$$

(Resistances R are estimated as for a heating system by totalling the number of bends, etc., and assuming an average size and velocity. Reading from the table at the bottom of Table LII an equivalent foot run for $R=1$ may be obtained.) Assume 10 long bends of $R=1$, size 2 in. average and velocity 3 ft./sec. (i.e. 9 ft. for

$$R=1).$$

$$\text{Equivalent } R = 10 \times 1 \times 9$$
$$= 90 \text{ ft.}$$

Therefore $T = 310 + 90 = 400$ ft. total T.

$$\frac{H}{T}\text{ per 100 ft.} = \frac{16}{4{\cdot}0} = 4{\cdot}0.$$

Reading from Table LII for $F = 4{\cdot}0$, we may now construct a schedule of deliveries as follows:

Size, Inches	G.P.M.
	1·0
	2·9
¼	11·4
1½	18·5
2	40
2½	73
3	119
4	260
5	440

from which the pipe sizes may be added to the diagram as shown.

Sizing Secondary Returns—Having established the sizes of the secondary flow pipes, it is a simple matter to calculate the heat emission from the circulating portions of these from Table XXXIII, which is given again below (Table LIII) for the particular temperature applying to hot-water supply systems, both for iron and copper pipes.

TABLE LIII

ISSION OF HOT-WATER SUPPLY PIPES AND FITTINGS IN B.T.U.'S PER HOU

Mean water at 150° F. Air at 60° F.

Nominal Internal Diameter, Inches	*B.T.U.'s per Hour per Lineal Foot—Bare Pipes: Galvanized Iron	Copper
½	48	30
¾	61	44
1	70	55
1¼	85	65
1½	96	75
2	113	96
2½	135	116
3	165	147
4	203	190
5	247	236
6	291	280

Linen Cupboard Coils:
2 in. dia., containing 18 ft. of pipe, say, 2000 B.T.U.'s/hr.

Towel Airers:
3 ft. × 3 ft. plated, 3 1¼-in. rails, say, 800 B.T.U.'s/hr.

To this must be added the losses from linen cupboard coils and towel airers, average emission for some examples being also given in the table.

* For lagged pipes divide these figures by 4.

The secondary returns must also carry sufficient water for their own heat emission so that the temperature drop back to the cylinder does not exceed a predetermined maximum such as 20° for a small system or 30° for a larger. Losses for these returns must therefore be added to each section, and since their size is not yet known they must be assumed at, say, one or two pipe sizes less than the flow in each case.

With the above totals marked on the plans for each branch or section of main it is possible to arrive at totals working back to the cylinder exactly as for a heating system.

The circulating pressure may be obtained as described on p. 192, by determining the average height from the centre of cylinder to average point of heat emission. The appropriate circulating head per foot of height may be taken from Table XXXVIII, allowing

flow 150°, return 120°, for large installation;
flow 140°, return 120°, for small installation.

The values will be found to be 0·103 and 0·067 in. per ft. of height respectively. Alternatively if a pump is necessary owing to the long runs or mains below the cylinder level, the head may be taken at 5 ft. for most systems, and 10 ft. or 15 ft. for large buildings or institutions with considerable distances between the blocks. The pump-head necessary may be found by allowing a maximum of 0·1 in. w.g. per foot of travel, allowing the total run of return pipe, plus one-third of the run of flow pipe (on account of its larger size) plus 25 per cent. allowance for bends and resistances.

Having determined the head, whether for gravity or pump circulation, the resistance of the flow mains (the sizes of which have already been determined) to the furthest point should be calculated as for the accurate sizing of a heating circuit. This figure, deducted from the total circulating head, will give the available pressure for the return mains. The latter divided by the feet run of return main including allowance for bends and single resistances, will give the available pressure per foot run.

From Table XXXIX a schedule may then be compiled for the pounds of water passed for the various sizes of pipe. The pounds multiplied by the temperature drop will give the B.T.U.'s transmitted for each size.

The sizes of the secondary returns for the transmissions already noted for each section may then be marked on the plans direct from this table, by taking the B.T.U.'s for each pipe as the maximum in each case.

It may so happen in certain cases that the return pipe may be found to be larger than the flow. This means that the circulation has determined the size rather than the outflow requirements. In such case the flow should be increased so as to be equal to the return. In general it will be found that the latter are smaller than the flows.

This method will not be entirely accurate, since by going over the sizes again now that their approximate diameters have been fixed it might be possible in some cases to reduce them. The nearer branches similarly might

be recalculated. It is very doubtful if this is ever worth while in the case of a hot-water supply system, since the saving in initial cost would be very slight and quite incommensurate with the time taken to do it. Furthermore, diameters on the large side are desirable when it is remembered that furring may reduce the internal area after a period of years. Indeed, if the water is known to be particularly hard it is worth considering the increase of the whole of the sizes, both flow and return, by one size throughout above that theoretically required.

An example of sizing of secondary returns is given below.

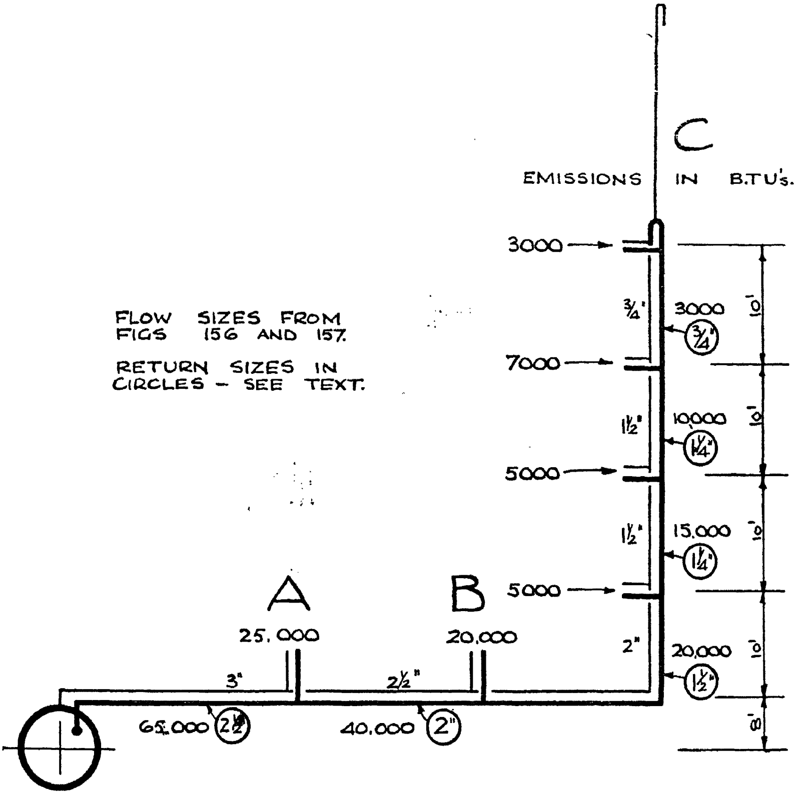


FIG. 158.—Up-feed System—Return Sizes.

Taking the system previously considered (see Fig. 157, p. 250), it is required to size the returns to riser C, using the method just described.

The emission of piping, towel airers and linen-cupboard coils for the various sections of pipe are taken out and marked on the diagram, as in Fig. 158.

$$\text{Average } H \text{ (say to } \tfrac{1}{3} \text{ height of riser)} = \tfrac{48}{3} = 16 \text{ ft.}$$

$$\therefore\ CP\ (150° - 120°) = 0{\cdot}103 \times 16 = 1{\cdot}65 \text{ in.}$$

Deduct resistance of flow-pipe:

Size	Length (of Flow) including R	B.T.U.'s	Lbs. for 30° Drop	Resistance per Foot, Inches	Total Resistance, Inches
3"	100	65,000	2170	0·0007	0·070
$2\frac{1}{2}$"	60	40,000	1330	0·0007	0·042
2"	80	20,000	670	0·0006	0·048
$1\frac{1}{2}$"	10	15,000	500	0·0015	0·015
$1\frac{1}{2}$"	10	10,000	330	0·0008	0·008
$\frac{3}{4}$"	15	3,000	100	0·0024	0·036
					0·219

Total resistance = 0·22 in.

Net CP for returns = 1·43 in.

T (returns only) including bends = 300 ft.

$$\therefore \frac{CP}{T} = \frac{1\cdot43}{300} = 0\cdot004 \text{ inches/ft.}$$

Hence B.T.U.'s transmitted are as follows:

Size	Lbs.	B.T.U.'s for 30° Drop
$\frac{3}{4}$"	131	3,930
1"	283	8,490
$1\frac{1}{4}$"	520	15,600
$1\frac{1}{2}$"	837	26,110
2"	1833	54,990
$2\frac{1}{2}$"	3330	99,900

from which the return sizes may be filled in on the diagram as indicated in circles.

It should be remembered that meticulous accuracy in the sizing of these service pipes is neither possible nor desirable, since the whole of the calculations are based on assumptions which may or may not be correct in practice. The best that can be achieved is to design the system on a reasonable and consistent method, such as that outlined above, which has been found in practice to give satisfactory results, provided due care is exercised in allowing for any special cases.

Draw-off through Returns—No account has been taken in the example of water flowing to the taps from the return pipe, which occurs where secondary circulation is used without a pump. In Fig. 159, when a tap is opened at X, water will obviously flow to it in both directions from the cylinder, the water in the return pipe flowing in the opposite direction from usual owing to the gravity circulation being completely overcome by the much greater head from the tank to the tap.

The reduced frictional resistance brought about in the out-flowing system, due to the double pipe, will vary inversely as the distance of any

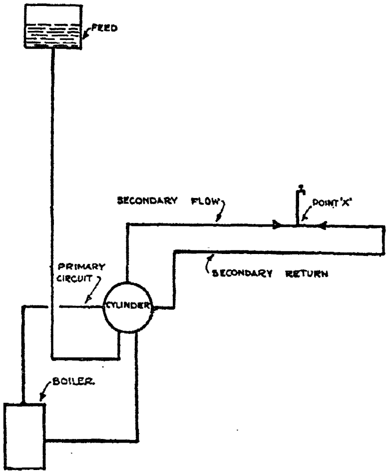


FIG. 159.—Showing Flow of Water in Secondary Circulation when a Tap is Opened.

tap from the cylinder. This effect is usually ignored when sizing pipes for small systems, but on a large 'ring-main' layout it should be taken into account.

DROP SYSTEM

A second method of secondary circulation is as shown in Fig. 160. In this case the flow riser delivers direct to a hot-water head tank above the level of the topmost tap, and the supplies to the taps are taken from a system of drops connected from the bottom of the tank.

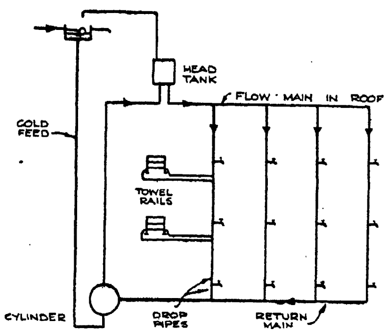


FIG. 160.—Hot-Water Supply Drop System.

The advantage of this method over the up-feed in the case of high buildings, is that the pipe sizes are generally less. The upper floors are fed downwards from the top tank or partly downwards and partly upwards.

As a result a greater height (H) may be taken from the cold tank to the

top tap than in the case of the up-feed system. An assumption that the top $\frac{1}{3}$ of the drops are fed downwards is generally reasonable. The remaining $\frac{2}{3}$ of the height is fed from the returns working in the reverse direction.

If the system has pump circulation with the pump in the return, obviously the whole discharge from the taps comes from the top. If the pump is in the flow the part upward-part downward delivery will be possible. Where the whole delivery comes from the top via the head tank and flow riser, the sizing of the drops and top flow mains is simplified by assuming that the head tank is in effect the cold feed tank. If the drops are vertical, it will be found that at each floor the $\frac{H}{T}$ is the same at each branch.

The disadvantages of the drop system are:

(i) That it involves ranges of large piping at or near roof level, often difficult to accommodate.

(ii) That towel airers and linen cupboard coils connected to the drops operate by local gravity circulation only, so that there is not the same possibility of connecting them when at a distance from the drop.

UP-FEED HEAD TANK SYSTEM

In order to avoid mains at or near roof level necessary with the drop system, the up-feed method may be used with a separate hot-water head tank to each riser as shown in Fig. 161.

At periods of heavy draw-off on the lower floors of any particular riser, when the supply to the top floor might tend to be reduced, this tank will supply the riser downwards by drawing on its storage, so ensuring a good

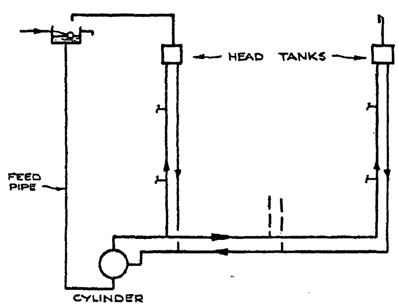


FIG. 161.—Hot-Water Service System with Separate Head Tanks.

supply at all levels. As it is unlikely that the heavy draw-off continues for more than a minute or so, the tank will not have emptied before the demand ceases, when the level in the system will balance up again by water flowing up the riser in the normal way.

The sizing of this system may be simplified by ignoring the top one or two floors and sizing flow mains and risers for the discharge up to the next floor down. The height H is thus greater than with a plain up-feed system and smaller pipes will result.

Sizing of Head Tanks—The size of the head tank can be determined by allowing two or three minutes' capacity for the discharge which is assumed to come from it. Thus if 25 galls. per minute come downwards in a drop system, and 50 galls. upwards, a head tank of $25 \times 3 = 75$ galls. would be adequate.

This storage should not be taken in full as reducing the main cylinder capacity, though approximately half its volume may be allowed as being effective.

Pump in Return Main—In the case of an accelerated secondary circulation with the pump in the return, the size of this unit may be determined from the emission B.T.U.'s as for a heating system.

For example, if the total emission is 600,000 B.T.U.'s per hour, and a 20° drop has been assumed, the capacity of the pump will be

$$\frac{600{,}000}{20 \times 10 \times 60} = 50 \text{ galls. per minute.}$$

Allowing a margin of say 20 per cent., a suitable size pump would be 60 galls. per minute. The head of the pump will be arrived at as discussed on p. 207.

Pump in Flow Main—Where a number of scattered buildings are served from a central source and the feed tank provides only a poor head in the highest blocks, it is possible to augment this by placing the circulating pump in the flow main. In this case the pump must have a capacity equal to the maximum discharge from the system, and the head will be determined by the augmented level required, plus pipe friction.

The placing of the pump in the flow main allows of the returns being used as delivery pipes as previously mentioned. A regulating valve may be necessary in the main return.

As for the heating systems, pumps of centrifugal type are most suited for the purpose, and again an automatic by-pass valve is desirable for gravity circulation when the pump is stopped. The pump body should be of a type readily opened for inspection and for removal of scale, and where the water is corrosive should be of gun-metal.

GENERAL

Vent and Feed Pipes—Vent pipes from the top of each riser carried above the feed tank are necessary unless it can be so arranged that the topmost tap acts as a vent when it is opened. The disadvantage with the latter method is a tendency to spluttering at the tap until the air is released.

It should be noted that the cold feed pipe has to furnish the whole supply, and it is a great mistake to pinch it for size. Even in the smallest installation 1 in. diameter should be the minimum size for this pipe.

Materials for Pipes—The choice of materials for hot-water pipes will depend on the water, which is really a separate subject. Galvanized iron is the cheapest that may be considered, as bare black iron pipes would cause discoloration of the water in most cases. It must be remembered, however, that the galvanizing is damaged whenever a pipe is cut or bent, and frequently it is only the deposit of fur which eventually protects these portions. Galvanizing after bending is often specified for the best work.

Copper piping is undoubtedly worth while wherever its cost can be met, owing to its permanence and cleanness of bore, which reduces furring. With the increasing facility of coupling brought about by new methods, its use has become more common. In soft water districts copper pipes and cylinders are in any event generally necessary.

With certain waters of the acid type, tinned copper is advocated, though a chemist's report is most desirable on any specific case, as sometimes tinning may do more harm than good.

Insulation—With a hot-water supply system it is most desirable that every foot of circulating pipe should be insulated, as the losses are going on twenty-four hours a day summer and winter, and may mount to a considerable figure. The heat emitted from such pipes if not insulated is also objectionable from the point of view of the temperature in the building in hot weather.

Such insulation may take the form of any of the non-conducting coverings already given in detail for heating systems. It is needless to say, of course, that the boiler and cylinder should also be adequately lagged, unless the latter is used for the warming of a linen cupboard in a small domestic system.

Valves—Valves on hot-water supply systems often give trouble due to fur or scale collecting on the faces, which in time renders them useless for shutting off the water. For this reason plug cocks are much to be preferred, especially if of the lubricated type. These can be relied upon to shut off tight even with the most severe internal encrustation.

Water-Softening is advocated for large systems wherever the water is hard, so as to avoid trouble with furring. The simplest system is the base-exchange, which uses the property of Zeolite to change the calcium salts into sodium salts. The process may be repeated indefinitely by regeneration with common salt. Another system is the lime-soda, which is cheaper to run but less easy to handle. This also is a separate subject in itself and cannot be dealt with in detail here.

CHAPTER XII

Heating and Hot-Water Supply by Gas

TOWN'S GAS

Town gas is obtained from the destructive distillation of coal in closed retorts which leaves a residue of coke, tar and ammoniacal liquor being obtained as by-products. Under the Gas Regulation Act 1920, gas undertakings are compelled to supply gas at a declared calorific value. In practice it is found that the gas from the carbonization of coal is of a higher calorific value than is necessary for this purpose, and it is usually diluted by addition of blue water gas (obtained from the gasification of coke) or by carburetted water gas (in which the blue water gas is enriched by the 'cracking' of oil at high temperature).

A diagrammatic layout of the plant in a gasworks is given in Fig. 162.

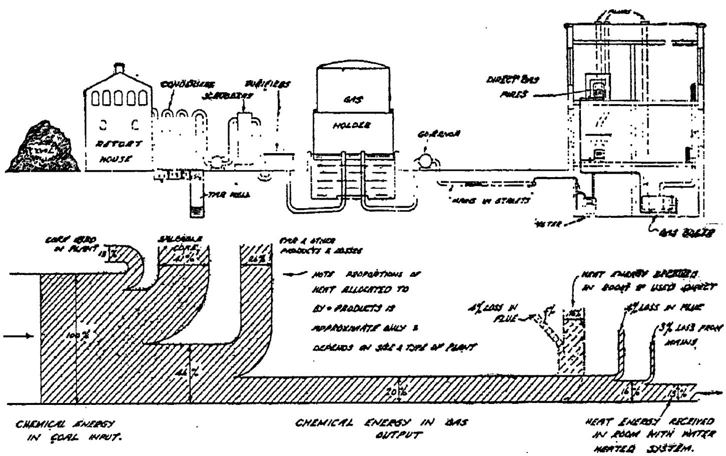


FIG. 162.—Diagram of Transmission of Heat by Gas.

A typical analysis of town's gas having a calorific value of 500 B.T.U.'s per cu. ft. is:

Carbon dioxide	(CO_2)	3·2%	by volume
Carbon monoxide	(CO)	15·0%	,,
Methane - -	(CH_4)	21·8%	,,
Hydrogen - -	(H_2)	46·9%	,,
Unsaturated hydrocarbons	(C_nH_m assumed composition C_4H_8)	3·0%	,,
Nitrogen - -	(N_2)	9·0%	,,
Oxygen - - -	(O_2)		
Specific gravity (Air = 1)	=	·48%	

F.H.

General Considerations—Gas, unlike electricity, can be stored in gas holders for use by consumers at any time. The manufacturing plant is run at constant load, and peak demands are taken care of by the storage capacity. Its use, therefore, is not restricted by considerations of maximum demand as in the case of electricity, where the generating plant has to be run at all times in accordance with the demands made by the consumer.

Gas is usually distributed to consumers at a pressure of 2 to 6 inches water gauge, and its flow can be varied infinitely between 'off' and 'on' by the operation of a cock or valve.

As a form of heating it possesses a number of distinct advantages: the absence of ash and smoke, its great flexibility, ease of control by automatic thermostats, etc., cleanliness, labour saving and reliability.

Taking into account the efficiency of the various processes of manufacture and the thermal efficiency of utilization of the gas, carbonization results in the greatest use of the potential thermal and chemical energy in the coal being made. Reference to Fig. 162 will show this.

Economic Considerations—Gas is a high-grade manufactured fuel and its cost is naturally higher than the other raw fuels.

A comparison of the process and losses between the use of gas or solid fuel for heating purposes can be appreciated from Figs. 162 and 163. The

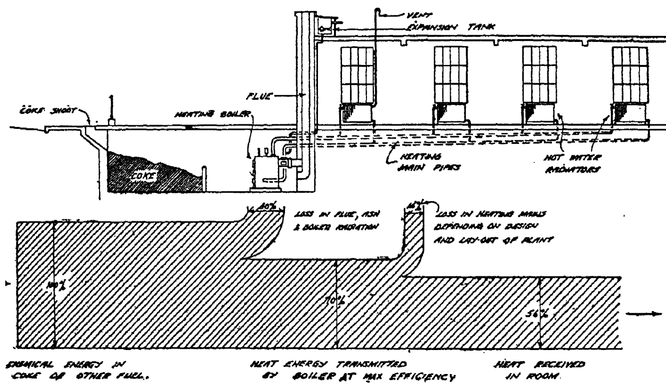


FIG. 163.—Diagram of Transmission of Heat by Solid Fuel.

relative costs of various fuels are given in Table XV (p. 51). In considering the relative figures given in this table the following points must, however, be borne in mind.

(*a*) The price of gas varies considerably according to the policy of the gas company supplying the particular area. In the north, where a gas grid has been developed in conjunction with the coke ovens for the surrounding steelworks, prices as low as 2d. per therm are quoted for large consumers. In London and other places two-part tariffs have been developed, where rates as low as 4d. per therm are quoted for large quantities.

(*b*) Gas is supplied at constant quality as regards calorific value. This is of great value in its use as a fuel, as the original adjustments of burners maintain constant air fuel ratios and ensure that the appliances are always working under conditions of maximum efficiency.

(*c*) With automatic control the heat supplied is in strict proportion to the heat required. This is not usually the case with solid fuels hand-fired, though it becomes nearly so with stoker-fired, oil-fired or magazine boilers. Some saving with gas should thus be expected, depending on which alternative system is being considered, but it is difficult to give an exact value to it.

The saving with an intermittent system, such as for a church or school, will be more than with a continuously operated one, such as a hospital or factory working twenty-four hours a day.

If gas is available at low rates for bulk heating, such as 4d. a therm, taking labour into account, it will be found to compare favourably with other fuels, and the case should receive close examination.

Calorific Value and Measurement of Gas—The calorific value is defined as the 'number of British Thermal Units (gross) produced by the combustion of 1 cubic foot of the gas measured at 60° F. under a pressure of 30″ of mercury and saturated with water vapour'. This declared value must be maintained within a limit of 5 per cent.

Gas is sold to the consumer on the basis of the therm, which is equivalent to 100,000 B.T.U.'s. The gas supplied to the consumer is measured by volumetric meter. The volume consumed is noted and the therms charged for in the following way:

$$\text{Therms consumed} = \frac{\text{Cubic feet consumed} \times \text{declared calorific value}}{100{,}000}.$$

Declared calorific values vary between 350 and 560 B.T.U.'s per cubic foot, and in one case a figure of 200 is used.

A common figure is 500 B.T.U.'s per cubic foot, and on this basis:

$$1000 \text{ cu. ft.} = \frac{1000 \times 500}{100{,}000} = 5 \text{ therms.}$$

HEATING BY GAS

Gas can be used for heating either indirectly in a hot-water or steam boiler connected to a system of pipes, radiators, etc., or directly by the use of local fires, convectors, radiant heaters or radiators.

INDIRECT HEATING BY GAS

Various gas boilers are shown in Figs. 164-167. These can be connected to any heating system in the same way as any other type of boiler.

Figs. 164 and 165 show two types of cast iron sectional boiler. The maximum head under which they work, life and other characteristics are

almost exactly similar to the solid fuel cast iron sectional type. The outputs vary from 22,000 to 1,400,000 B.T.U.'s per hour. The boilers may be used for hot-water heating and indirect hot-water supply in the usual way,

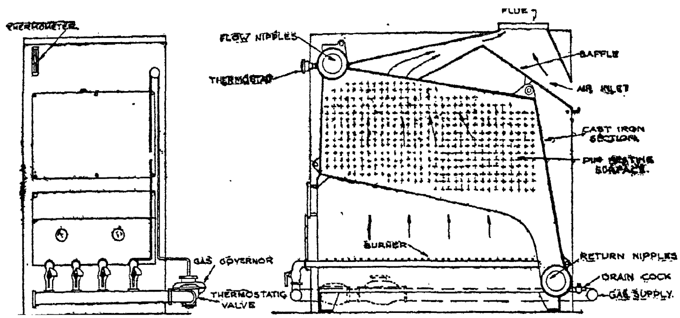


FIG. 164.—Cast-Iron Sectional Gas Boiler ('Ideal').

although the water ways of the boiler shown in Fig. 165 may be used for direct hot-water supply.

The gas is burnt by means of a number of flat flame burners supplied with gas at a constant pressure through a governor, in order to maintain high efficiency and give complete combustion.

Thermostatic control is provided of the relay-operated type, which varies the gas flow according to the temperature of the flow water. The jets always remain alight on a low flame.

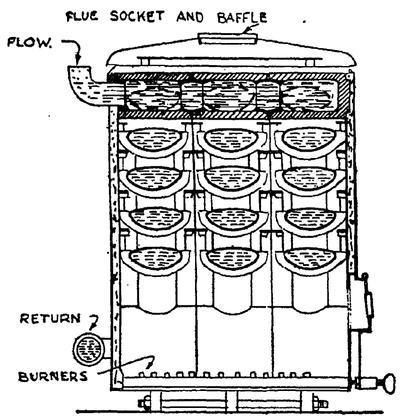


FIG. 165.—Cast-Iron Sectional Gas Boiler ('Rex').

To obtain efficient combustion with gas it is necessary that the draught conditions at the point of combustion should be correct and, as is well known, flames are very sensitive to small changes in pressure. Owing also to the low temperature of the flue gases in gas boilers (on account of their high efficiency), there is a greater tendency for down draughts to occur in chimneys and flues to which they are connected.

Gas boilers are, therefore, fitted with bafflers or draught diverters before the flue gases enter the flue pipe or chimney.

This allows free removal of products of combustion and diversion of any down draught away from the combustion chamber. Down draughts cause incomplete combustion with the formation of carbon monoxide, and

may cause an explosive mixture in the boiler. With the two boilers shown in Figs. 164 and 165 the draught diverters are made integral with the boiler.

Sulphur is present in gas to the extent of 25-50 grains per 100 cu. ft., and on combustion this causes a certain amount of corrosion on water surfaces, etc. Gas apparatus, therefore, has to be cleaned at periodical intervals. The efficiencies of the two types of boilers illustrated in Figs. 164 and 165 is between 80 and 85 per cent.

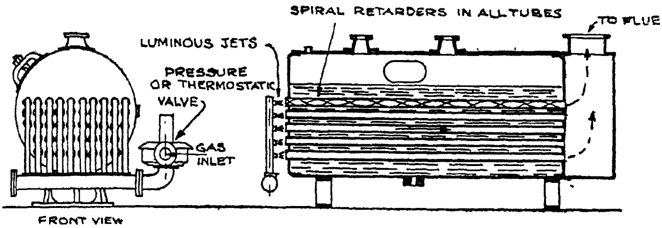


FIG. 166.—Horizontal Gas Boiler ('Town's Gas' or Bonecourt).

Figs. 166 and 167 show two types of riveted steel gas boilers with fire tube heating surface. Either is suitable for water heating or for high pressure steam, on account of which they are frequently used for giving steam supplies to kitchens, laundries, printing works and other places where a small supply of high-pressure steam is necessary and where space is a serious consideration.

Fig. 166 has plain tubes with spiral metal 'retarders' in each. These serve to produce a higher transmission from the heating surface due to the swirling effect on the gases. The burners are luminous, and when the thermostatic or pressure-stat control cuts down the gas, these jets always remain alight as small flames, which are not then pulled through the tubes but burn upwards into the atmosphere. This makes the boiler-house hot if it is in a confined space, and the heat so produced is wasted. The working efficiency of this type is stated to be about 84 per cent.

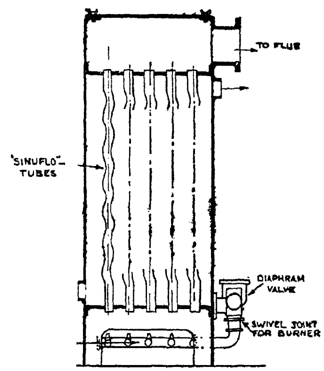
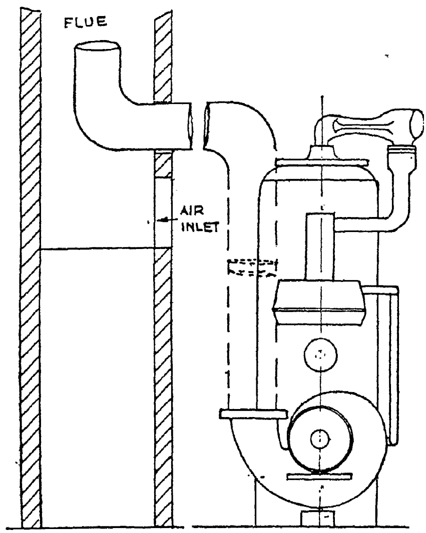


FIG. 167.—Vertical Gas Boiler ('Sinuflo') for Steam or Hot Water.

Fig. 167 has tubes of the special wave form already mentioned on p. 99, again with the object of raising the transmission per foot run over that of plain tubes. The tubes are stated to be $2\frac{1}{2}$ times as effective per foot of length as plain tubes.

The burners and controls are similar to the type shown in Fig. 166, but

as the boiler is vertical no waste occurs on minimum flame. The efficiency is about the same for both types, and each ranges in size from about 100,000 to 2,000,000 B.T.U.'s per hour.



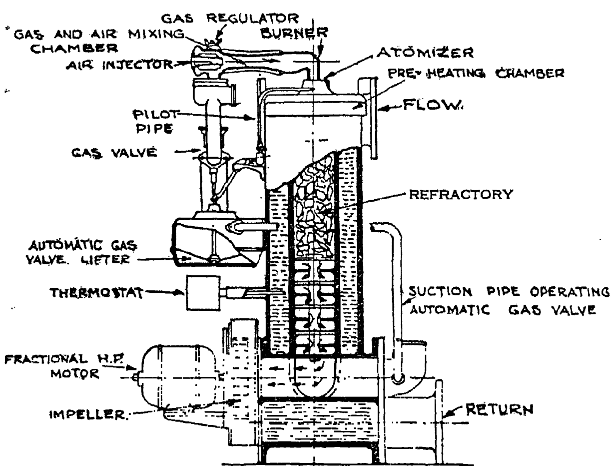


FIG. 167 (*a*).—High Efficiency Gas Boiler ('Vesta').

A highly efficient gas boiler is the 'Vesta' (Fig. 167 (*a*)). In this there is a fan drawing air through the combustion chamber for combustion of the gas which impinges on to a silica refractory material enclosed by water-

ways. Very high temperatures are obtained in the combustion chamber with resultant high heat transfer rates, and the temperature of the flue gases is reduced to within 20° F. of that of the return water. Efficiencies as high as 92 per cent. have been obtained with this boiler.

For domestic, household and small industrial loads, etc., use can be made of one of the small gas circulators. A typical gas circulator is shown in Fig. 168. This is constructed with iron or copper waterways, and may be

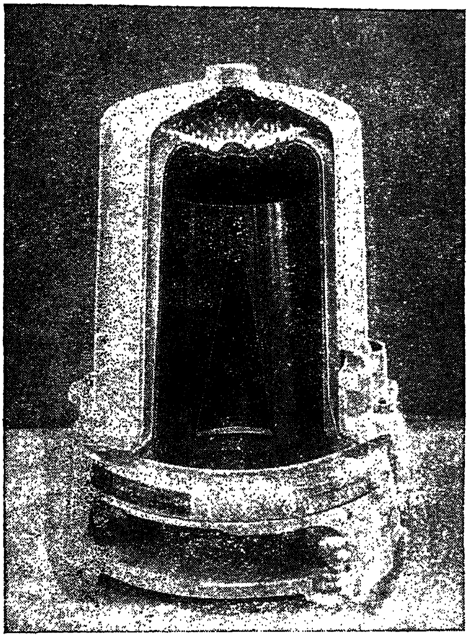


Fig. 168.—'Ironclad' Gas Circulator (Richmond Gas Stove Co.).

connected direct to any heating or hot-water supply system. The circulator can be used for direct hot-water supply and, with soft waters, copper waterways are used. Otherwise iron waterways are sufficient. A gas burner, which may either be of the flat flame or bunsen type, is controlled by a direct-acting thermostat placed in the return pipe, with a cylinder for direct hot-water supply.

These appliances may be used in batteries for larger loads. They are obtainable in sizes from 25,000 B.T.U.'s to 45,000 B.T.U.'s per hour. Their efficiency is of the order of 75 per cent. With this type draught diverters must be fixed on the flue pipes.

For conversions of existing boilers to gas burning or for special applications, a number of gas-burner systems are available. Summarizing briefly, these can be divided into:

(*a*) *Atmospheric air low-pressure gas type.*

There are a number of ingenious designs to get efficient mixing of the gas and air and regulation of the air supply. They can be easily applied to conversions from solid fuel type boilers, and thermostatic control can be easily adapted to the requirements of any particular case.

(*b*) *Air blast and low-pressure gas.*

In this burner the full amount of air for complete combustion is supplied by means of a small fan. The gas-air mixture burns on a refractory cylinder or bed, giving a large proportion of its heat as high temperature radiant heat. The gas is entrained by a Venturi system which is automatic in action, so that as soon as the air is flowing the gas valve opens, and the gas-air mixture goes to the burner to be burnt. High efficiency can be obtained and thermostatic control can be applied. The system is illustrated in Fig. 169.

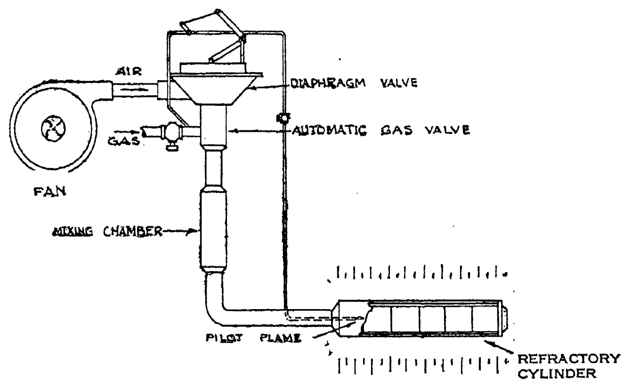


FIG. 169.—Radiant Gas Firing Unit (Cox's).

(*c*) *High-pressure gas atmospheric air.*

In this type of burner the full amount of air for combustion is entrained by the injector due to the high velocity of the gas issuing from the nozzle. The gas-air mixture is burnt on refractory material, and gives high proportion of radiant heat. Thermostatic control can be applied to the burner.

Conversions can be made to suit the load on any boiler. Their efficiency is generally lower than a specially designed gas boiler, and the results obtained will depend on the conditions applying in each individual case.

There is also a system of *submerged combustion* which has been applied to heating. In this the air-gas mixture is burnt in the water to be heated, and an efficiency of nearly 100 per cent. is attained. However, the water dissolves the oxides of sulphur and some of the carbon dioxide, and results in it becoming corrosive. On this account this system cannot be recommended for a heating system.

DIRECT HEATING BY GAS

Gas is used for space heating by the use of gas fires, convectors, radiators and radiant heaters. These appliances can be classified under two headings, viz. those for local heating and those for general space heating.

Local Heating—*Gas Fires*—One of the most popular forms of local gas heaters is the gas fire.

Considerable research has been carried out on gas fires, and as a result the 'Highbeam' was developed some years ago, having a radiant efficiency

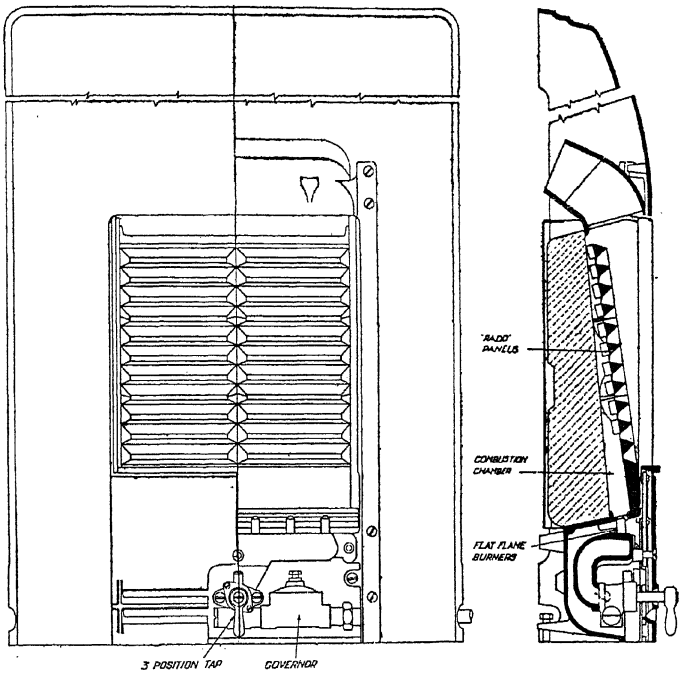


FIG. 169 (*a*).—Diagram showing Construction of the 'Silent Beam' Fire.

of 54 per cent. on the gross calorific value of the gas. Recently further research has produced the **'Silent Beam'**, which is an improvement on the above, with flat flame instead of 'Bunsen' type burners.

The 'Silent Beam' fire will probably replace the older type, so that a few notes on its characteristics are given.

The radiants are in the form of 'louvres' in three sections (see Fig. 169 (*a*)) in place of the columnar type. They are slightly inclined backwards, and though their radiant efficiency under test conditions is now less at 45 per cent., their actual working efficiency as fitted in a room is stated to be the same as that of the 'Highbeam'.

The fire is illustrated in Plate XV (*a*)*, and it will be seen there is no

* Facing p. 274.

canopy. The design is such that the radiants are fully heated to the top, and ventilation air from the room previously drawn in at the canopy is now extracted through slots at the sides at a rate equivalent to four changes per hour in a room of average size.

The flat flame burners give silent operation. There is the usual spark ignition. A governor is fitted to control the gas supply, and is adjustable for various qualities of gas from 375 to 525 B.T.U.'s per cubic foot. There is nothing to get out of order or collect dust as in the old type of fires, and once installed they should give no trouble.

The gas fire is a useful method for quickly radiant warmth for intermittent use. It is not an efficient method of heating a room continuously. It is essential that a flue be provided either in the form of a chimney or by one of the hollow block type flues.

Various appliances have been developed to utilize the waste heat in the flue gases. One of these is the 'Raytonic' fire illustrated in Plate XV (*b*). In this fire radiant heat is obtained from the radiants and convected heat is recovered in the air passage through which the air of the room circulates.

Flueless Heaters—With regard to the numerous flueless gas heaters, including radiators, convectors, etc., their efficiency is, of course, 100 per cent. of the net calorific value of the gas, as the whole of the heat is liberated in the room, but care must be taken in their use as the products of combustion are given off into the air to be breathed. In connection with this the summary of the Report* of a Medical Advisory Committee is quoted as follows:

'The committee has examined all the available evidence on the ventilation of flueless rooms and the warming of them by flueless heaters. Since the air change in such rooms may be as small as 1 per hour, it is open to serious doubt whether a degree of ventilation of this low order can be regarded as satisfactory whatever the nature of the heating, and whether this modern tendency in housing construction is, from the point of view of health and comfort, in the best interests of the community. More information is needed on this subject.

'Any form of heating in small unventilated rooms which tends to lead to accumulations of hot stagnant air is objectionable. In this respect flueless gas heaters are no better and no worse than alternative appliances. Any risk to health from the inhalation of products of combustion from flueless gas heaters is discounted by the fact that, even with an air change as low as 1 per hour, discomfort from overheating would be experienced long before harmful concentrations of such products would be reached.'

Apart from the health aspect, it should be remembered that the sulphur in gas fumes is damaging to leather, fabrics, and to tarnishable metal work.

Space Heating—For small space heating, such as halls, corridors, small

* *Brit. Med. Journal*, August 10, 1935, vol. ii, p. 268.

restaurants, etc., convectors, radiators, etc., are suitable. These are of various types and designs to suit the purpose for which they are required. They can be controlled by thermostat individually or in groups. A typical modern convector, the 'Luma', is shown in Plate XV (*c*).

For large space heating, such as in factories, etc., use may be made of gas radiant heaters, gas unit heaters or gas heated plenum heaters.

Radiant Heaters have been developed which do not heat the air space as a whole, but radiate heat from a high temperature surface on to the occupied area. By this means comfortable conditions are attained with a smaller quantity of heat.

One such type, shown in Fig. 169 (*b*), is luminous and works with high-pressure gas. Sufficient air is entrained for complete combustion, and the air-gas mixture passes to a porous refractory brick. The mixture passes through the brick and is burnt on the surface, and the brick attains a bright red heat. It is fixed at an angle overhead to direct the radiant heat where required.

A non-luminous type using low-pressure gas is shown in Plate XVI (*a*). Being silent in operation, it may be used, for example, in a church. The heat is radiated from a steel black-enamelled surface, behind which there are a number of flat flame burners with a refractory insulating brick behind the flames. This reflects the heat on to the enamelled steel plate. The heaters are made in a number of shapes, bowl, wedge, wall type, etc., according to the direction in which the heat is required. The temperature of the radiant heating surface is not so high as with the high-pressure type, being about 500° to 600° F. Each heater is provided with a governor to ensure correct combustion by the burners, and can be remote controlled.

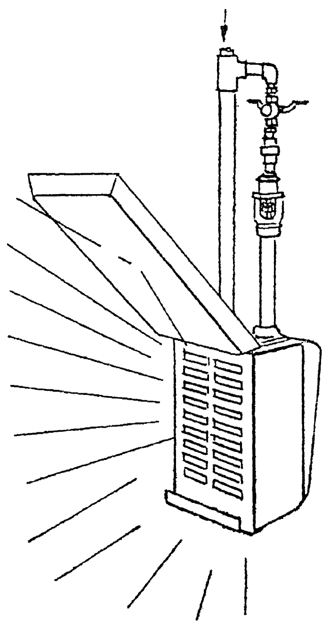


FIG. 169 (*b*).—High-Pressure Gas Luminous Radiant Heater (Radiant Heating Ltd.).

Both the high-pressure and the low-pressure heaters can be thermostatically controlled if it is so desired.

Air Heaters—A gas-fired unit heater is shown in Plate XVI (*b*) (facing p. 275). This is suitable for factories or public garages where no supply of steam or hot water is available for other purposes, and where space for a boiler or the labour attaching thereto is troublesome. The fan is started or stopped under thermostatic control, the gas supply being regulated similarly, and the products of combustion are discharged through a flue to the roof.

Where a plenum system of heating is adopted (as in factories, etc.),

the warming of the air may be accomplished with a gas heater of the type shown in Fig. 170. The cost of gas is the controlling factor here, unless space for a boiler rules other methods out. So far very few cases appear to exist where this type of heater has been used for warming ventilation air, though it is applied extensively to drying and other industrial processes.

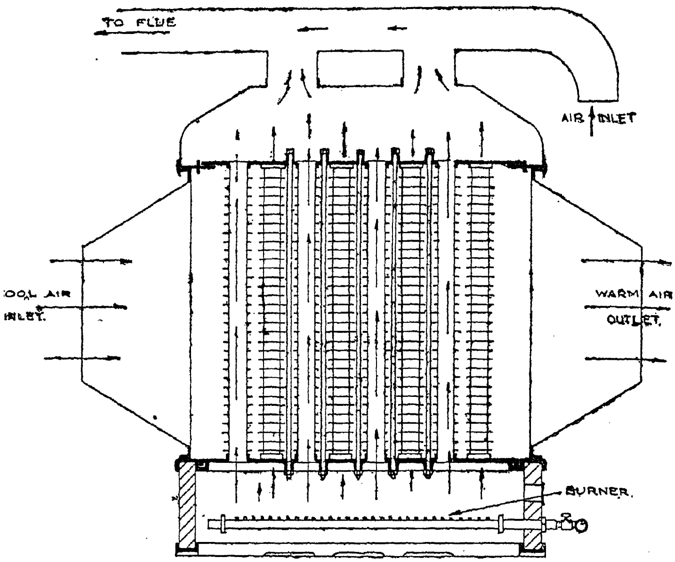


FIG. 170.—Gas-Fired Air Heater.

In America gas has been successfully adopted directly for Projection type unit heaters of the type shown in Fig. 41 (*c*) for heating large areas.

Sizing of Gas Heaters

(*a*) *Convection Heaters*, i.e. convectors, radiators, etc., unit heaters. Gas consumption is calculated direct from heat losses of the room or building with suitable margin for quick heating up. This has to be adjusted for apparatus where a flue conducts waste gases to outside by applying an efficiency factor. The B.T.U.'s divided by the calorific value per cubic foot gives the gas rate. The appropriate margin may be taken from Table XVI, p. 66. The number of heaters is then determined by dividing the gas rate calculated as above by the gas rate per heater, according to the type and size selected to give reasonable distribution.

(*b*) *Radiant Heaters. Luminous and non-luminous overhead types.*—Gas consumption is calculated from the heat losses as for convectors, except that a lower air temperature is assumed on account of the directional radiant effect on the occupants. It will be appreciated from the discussion on

Equivalent Temperature (p. 38) that for this to be constant in a room having high radiant heat incident on the occupants, the air and surrounding surfaces will be at a temperature lower than that with convected heating. This reduction is normally assumed to be 5°; i.e. for a room in which 60° F. temperature is required, the heat input is calculated on heat losses based on 55° F. To this an allowance for intermittent heating should be added, given by the makers as about 25 per cent.

(*c*) *Gas Fires.*—The estimation of the size of a gas fire from the heat losses of the room in which it is placed is of doubtful value. When a gas fire is set going in a cold room it is possible to feel some degree of comfort within the circle of radiant heat with air temperatures in the thirties or forties. Further, these heaters are chiefly used for intermittent heating, so that stabilized conditions are rarely obtained.

It is, however, usual to estimate the gas consumption from the heat losses at 60° for a particular room, allowing for three to four changes of air per hour. The result is doubled to allow for intermittent heating, and doubled again to allow for inefficiency of the fire.

One radiant of the 'high beam' type burns $4\frac{1}{2}$ cu. ft. per hour, i.e. 2250 B.T.U.'s/hr. with gas of 500 C.V. This varies with other types of radiant.

$$\text{No. of radiants} = \frac{\text{Heat loss} \times 2 \times 2}{\text{B.T.U.'s/radiant}}.$$

The 'Silent Beam' standard size has a gas rate equal to 15,000 B.T.U.'s per hr.

Where a gas fire is used to boost the temperature of a room heated basically by other means, such as by hot water radiator, the size of fire can be arrived at as above, allowing, say, 10° rise in the heat losses to be catered for by gas.

HOT-WATER SUPPLY BY GAS

Central System—A gas boiler may take the place of a solid or liquid fuel boiler in a central system of hot-water supply with storage cylinder and distributing pipes, such as has already been described in Chapter X.

The deciding factor as to whether gas with its attendant advantages should be adopted is generally entirely one of cost and will not be referred to again, as the same considerations apply as with heating. It should be borne in mind, however, that solid fuel hot-water supply boilers are generally not so efficiently designed as those for heating, for reasons which have previously been discussed, so that the comparison with gas should often be slightly more favourable to the latter than in the case of heating.

Any of the types of gas boiler referred to above is suitable for hot-water supply, either with or without a calorifier, according to the design of boiler and properties of the water. In addition there are, however, a number of gas boilers specially designed for hot-water supply directly connected to a cylinder, as shown in Fig. 170 (*a*).

All types are, of course, easily controlled thermostatically from the storage water temperature, either by means of direct-acting thermostats or relay type, and such control is essential for maximum economy. In certain types of boiler the thermostatic control is of the direct type inserted in the flow connection to the boiler.

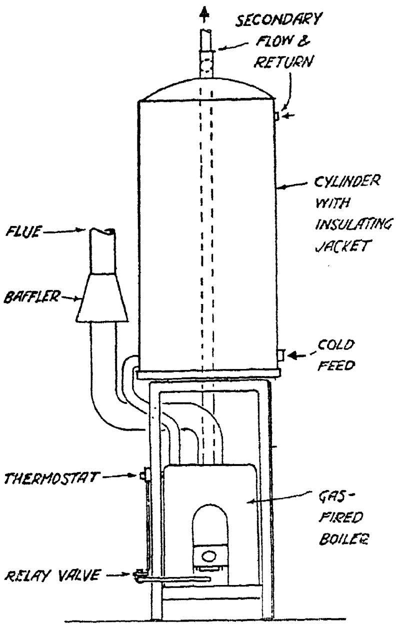


FIG. 170 (*a*).—Gas Boiler and Cylinder (Potterton's 'Empire').

Local Gas Heaters—The advantages of local heaters for hot-water supply have already been referred to in Chapter X (p. 222), and may be summarized as follows:

(*a*) Elimination of long runs and piping and the consequent loss by radiation.

(*b*) One piping system serves both hot and cold taps.

(*c*) Higher efficiency-ratio of heat put into water: heat delivered to taps.

In making reference to the lower cost of gas, it is only fair to state that in some cases where low rates for electric water heating are quoted gas has very little advantage in this respect.

Gas is able to meet sudden demands for large quantities of hot water better than electricity, since the loading of the latter always has to be kept at a minimum so as to reduce the 'maximum demand'.

Local gas hot-water heaters are of two main types, non-storage and storage.

(*a*) Non-storage Heaters

These are sub-divided into non-pressure and pressure types.

The non-pressure types comprise the well-known 'Geysers'. The water is under atmospheric pressure, and they are usually of the single-point type. While they are not free from serious disadvantages, they represent a convenient method of introducing a hot-water supply to a bathroom which lacks a better one. Care has to be taken to ensure adequate air inlet and an efficient flue system, as they are liable to become dangerous to occupants in unventilated bathrooms otherwise. There are a variety of designs available, and the modern geyser has an efficiency of 75-80 per cent. That most generally used is the closed type, i.e. the gas and water are kept separate. Efficiencies of 90 per cent. or over are obtained with the 'open' type,

where the heat is extracted directly by the water from the products of combustion. The use of the latter type of geyser is, however, less general than formerly, due to the contamination of the water.

The gas consumption varies from 180-240 cu. ft. per hour.

Various devices are incorporated in the designs of geysers to make them less dangerous.

They can be fitted with a tank for water supply or can be supplied from the cold water main, according to the regulations of the local water authority.

An improved instantaneous heater is the Pressure type or Multipoint, one make of which, the well-known 'Ascot', is shown in Fig. 171. They are designed to be instantaneous in action, so that when a water tap is turned on the gas lights and heats the water passing through the heater.

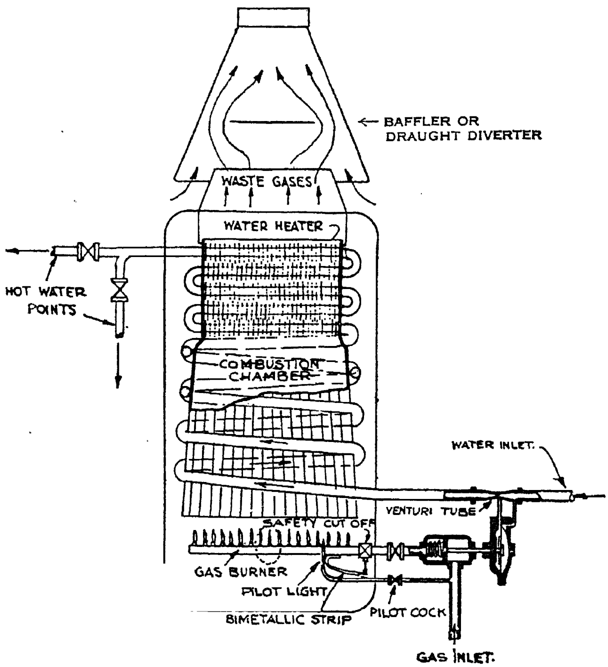


FIG. 171.—Diagrammatic Section of Instantaneous Multipoint Water Heater ('Ascot').

This may be connected to a number of taps, any one of which may draw from the same apparatus. The gas supply is always alight on a pilot flame, and the main burner is turned on automatically as soon as water is drawn off. The ingenious Venturi tube device operating a diaphragm gas valve will be noted from the diagram.

Such heaters may be connected from a tank supply or direct to the

main, depending on the pressure, but the former is preferable. A head of at least 12 ft. is required.

The rate of flow from this type of heater is necessarily restricted, and it is not to be assumed that a good supply may be obtained from more than one tap at the same time.

Whilst suitable for small domestic installations and a variety of similar uses, they are not to be advocated where sudden demands for large quantities of hot water are required at a number of points simultaneously. In such case a storage heater is necessary. The non-storage heater having no radiation loss during periods of non-draw-off has an advantage over the storage type in overall efficiency, particularly where the periods of draw-off are infrequent. As the demand for hot water becomes more continuous this advantage becomes less important.

These heaters have an efficiency of 80 per cent. when properly adjusted.

(*b*) Storage Heaters

These may be divided into low and high consumption types.

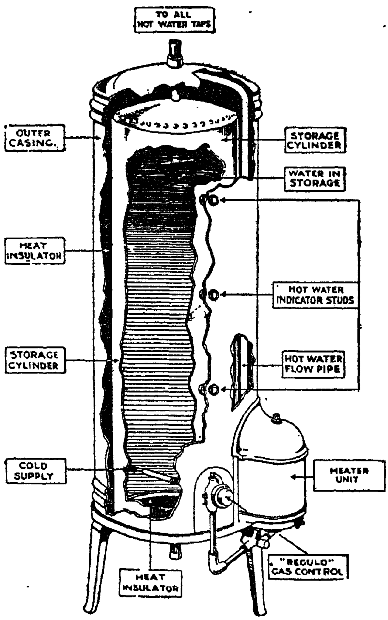
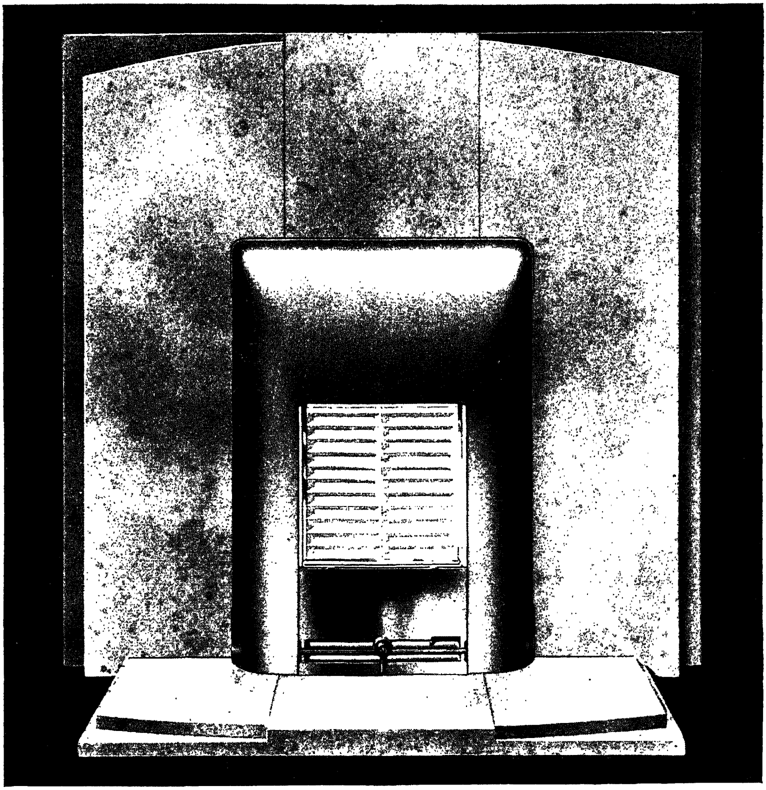
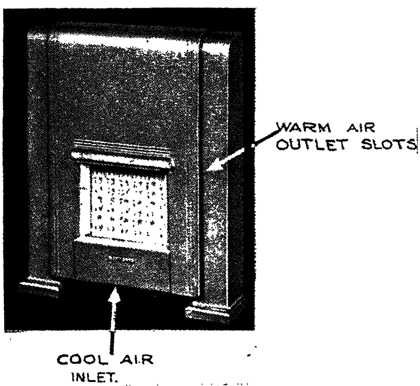


FIG. 172.—Gas Storage Heater ('Sunhot').
Low Consumption Type.

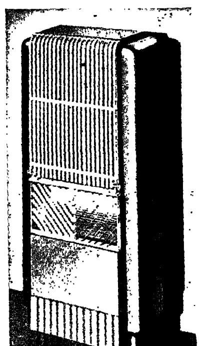
A typical low consumption type is shown in Fig. 172. Various capacities are available from 12 to 40 galls., and it may be connected to a hot-water supply system and any number of taps without, of course, any secondary circulation. It has a recovery rate of 4 galls. per hour raised 90° F. and a



(*a*) *'Silent Beam' gas fire*

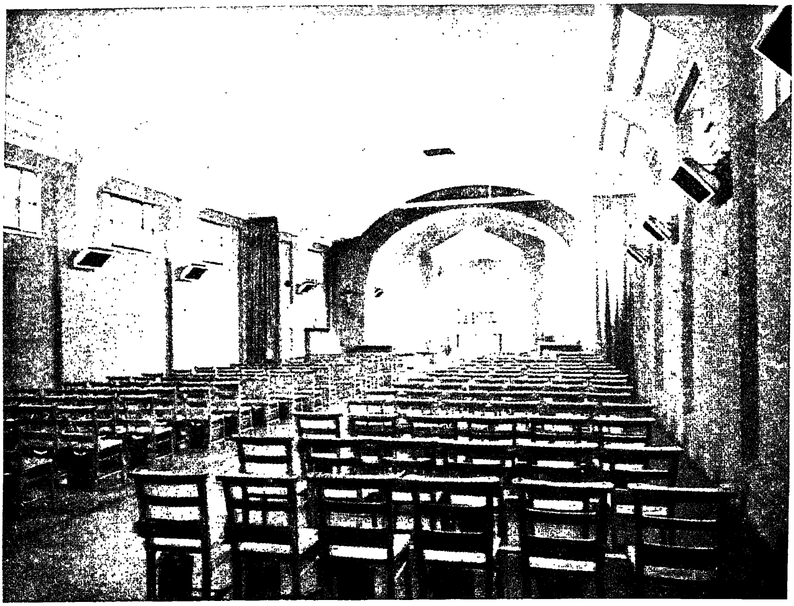


(*b*) *Gas fire* (*'Raytonic'*)

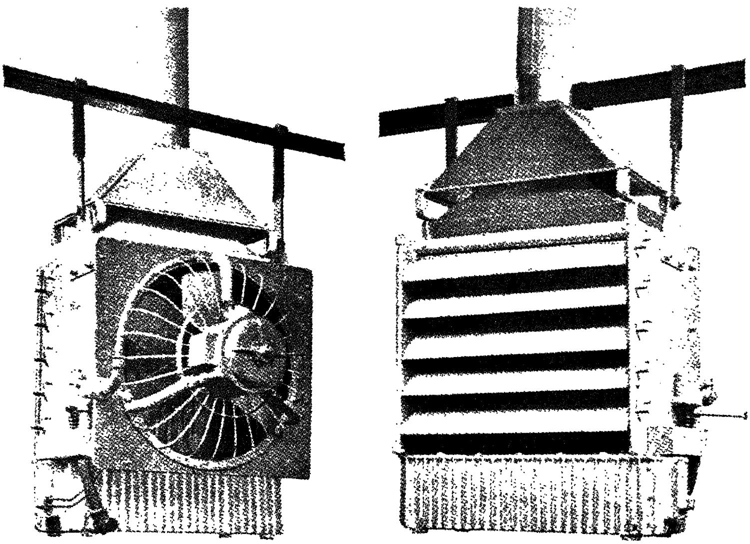


(*c*) *Gas Convector 'Luma'*
(*John Wright*)

Plate XV. Gas Heating Appliances (*see pp. 267-9*)



(a) Radiant Heaters (Bratt Colbran)



(b) Gas-fired unit air heater

Plate XVI. Gas-fired Space Heaters (see p. 269)

gas consumption of 10 cu. ft. per hour (500 B.T.U.'s gas). It has an efficiency of 75 per cent. when on full load. Having such a small consumption no flue is necessary.

High consumption types are similar in construction and fitted in a similar manner to hot-water supply systems, but the heating unit has a higher recovery rate. The cylinder capacities vary from 12-40 galls., and the recovery rate is 36 galls. per hour, raised 90° F. with a gas consump-

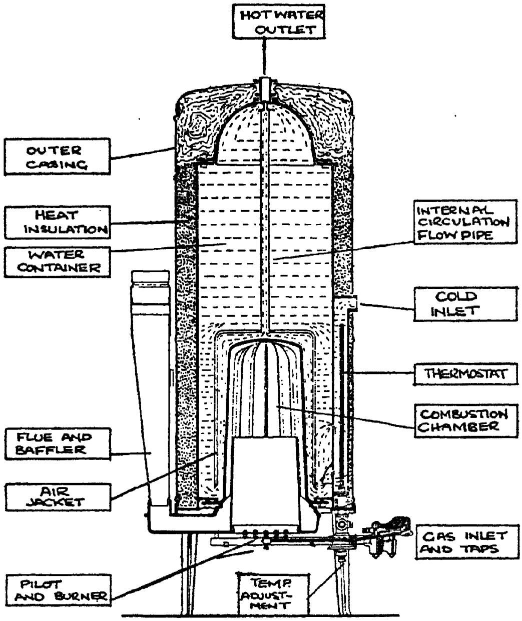


FIG. 172 (*a*).—Gas Storage Heater.
(Richmond's High Consumption Type.)

tion of 55 cu. ft. per hour (500 B.T.U.'s gas). A typical example is the 'Equator' heater (see Fig. 172 (*a*)). These heaters are thermostatically controlled. With the high consumption type of heater a flue is, of course, essential. The ball valve type, Fig. 140 (iv), may be used where a roof tank cannot be provided.

There are a variety of storage heaters of different types and makes which cannot come under the two headings above, and space does not allow for a comprehensive list and description to be included. They are, however, governed by the same principles of design and have the same salient characteristics.

(*c*) Miscellaneous Heaters

There are a great variety of makes and types of small single point sink heaters, both of the instantaneous and storage type. These are useful little

heaters which are cheap in first cost and in operation. Their efficiency is high, of the order of 80 per cent., and on account of their low gas consumption no flues are necessary.

One very interesting development and useful single point heater is the boiling water type. These small appliances will provide boiling water at the turn of a tap, and can be easily fitted over a sink. They have a high efficiency, 70 per cent. (compared with 40-50 per cent. by boiling a kettle on a ring), and need no flue. They will provide 2-3 pints of boiling water per minute. Care must be taken to install them in districts where the hardness of the water does not exceed 13° (Clark), as otherwise trouble will be experienced with 'furring up'.

In general, care must be taken with the water supply to which heaters are connected, and special metals used where the water is very soft, and periodical descaling carried out where the water is hard.

Thermostats for Gas Heaters—Thermostats for gas heaters may be of the direct or relay type.

There are several designs of the direct-acting type. One of the most reliable is the rod type, shown in Fig. 172 (*b*). This consists of a metal tube

(*a*) Expansion tube.
(*b*) Non-expanding rod.
(*c*) Valve.
(*d*) Spring.
(*e*) Adjustment.

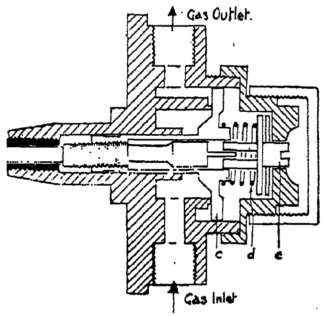


FIG. 172 (*b*).—Direct-Acting Gas Thermostat.

which expands with heat, to the inside of which is attached at one end a non-expanding metal rod. Movement of the rod due to variations in temperature of the outside tube is transmitted to a mushroom type valve, which is either raised or lowered on to a seating, thus increasing or decreasing the flow of gas to the burner. The thermostat is usually used direct in the water flow or return. A bye-pass screw is provided to keep the gas alight on the burner when the thermostat shuts down the gas supply. Other direct-acting thermostats utilize the expansion of a capsule containing a volatile liquid such as ether, for opening or closing a gas valve.

In the case of a relay type thermostat, a diagrammatic sketch of which is shown in Fig. 172 (*c*), a modified rod type thermostat is used, either in the room, or in the water flow or return. This varies the quantity of gas passing a small orifice and consequently the pressure, which passes over the top of a weighted diaphragm placed in the gas stream to the burner. As

the pressure increases on top of the diaphragm, becoming more nearly equal that on the bottom, the valve closes, until finally, when the thermostat is closed, the pressure on top equals the pressure underneath, and the weight of the diaphragm lowers it and shuts off the gas supply. An adjusting screw is provided to allow a certain amount of gas to pass to keep the burners alight. The small amount of gas passing the thermostat when it is open is piped to the burner where it burns.

Thermostatic control with gas gives a perfectly modulated action. Thus, the heat requirement at any particular time is in direct proportion

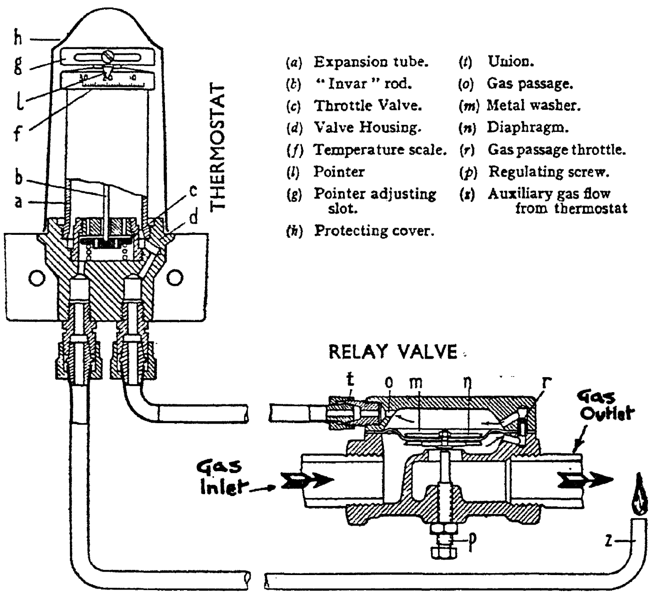


FIG. 172 (c).—Relay Type Gas Thermostat.
(Potterton's 'Perfecta'.)

to the amount of gas passing the thermostatic valve. This is a great advantage, and gas can be adapted to thermostat control more easily than any other fuel.

Gas Flues and Bafflers—The high proportion of hydrogen in gas produces copious condensation in the flue as the products of combustion cool down, this effect being the more serious with large systems and high efficiencies. Due to this, it is common practice to install gas flues in non-corrodible materials such as asbestos cement instead of iron; similarly it is desirable for the chimney to which such flue pipes are connected to be lined with a fire tile lining, preferably glazed inside. Such linings vary in thickness from $\frac{3}{4}$ in. to $1\frac{1}{2}$ in., according to the size of the flue, and are placed in sections vertically above one another, the ordinary brickwork being built round

them. If such a lining is not provided there is a danger of staining through the brickwork, particularly at the upper levels.

Below the point at which the flues enter a large opening is generally left, provided either with a plain grating or a draught stabilizer. The latter is particularly necessary where the draught is required to pull the gases through the flue tubes, as with the boiler shown in Fig. 166. Such an opening also serves as an explosion relief in the event of some fault occurring in the ignition.

As explained earlier in this chapter, gas heaters are very susceptible to changes in pressure and to down draughts on account of the low final exit temperature of the flue gases. Care must, therefore, be taken to see that flues discharge the products of combustion freely to the atmosphere. In buildings with return walls, areas, etc., peculiar atmospheric pressure conditions occur and, in order to get over this, it is always advisable to extend the flue to a height of at least 18 inches above the eaves of the roof. The possibilities of down draught are then much reduced. Cases are frequently seen where flues discharge directly to areas in an attempt to eliminate the cost of a long length of flue pipe. High external static pressures are usually obtained in cases like these and down draughts are experienced, and in some cases the static pressure is so high that air flows from the outside down the flue to the heater, the products of combustion all being discharged into the room.

Every flue should be fitted with a terminal of some kind to prevent birds nesting, etc. A good terminal will ensure the maximum efficiency being obtained from a flue.

Bafflers, or more correctly draught diverters (see Fig. 171), should be fitted to all gas heaters if these are not incorporated in the design of the heater (as in the case of some water heaters, boilers and gas fires). The perfect baffler has not yet been invented, but there are many good designs available. The use of a baffler takes care of any intermittent 'downflow', and maintains the efficiency of the heater even with excessive updraught.

GAS SUPPLY PIPES AND OTHER CONSIDERATIONS

The gas volume required by any gas boiler or gas heater can be easily calculated from

$$\text{Cu. ft. per hour} = \frac{\text{B.T.U.'s per hour}}{\text{calorific value of gas per cu. ft.}} \times \frac{100}{\text{percentage efficiency}}.$$

It is usual, however, for manufacturers of gas heaters to give the gas rates of the apparatus they manufacture, and from this the volume of gas required to be delivered to any heater can be found.

The delivery of gas through piping is given in Table LIV. This table is

based on Mr. Stephen Lacey's paper before the Institution of Gas Engineers, June 1923. It assumes gas having a specific gravity of 0·5 (air = 1), and is based on one-tenth inch water gauge loss in pressure between the ends of the pipe.

TABLE LIV

Sizing of Gas Pipes

Flows in Cu. Ft. per hour corresponding to $\frac{1}{10}$ inch W.G. Loss of Pressure between the ends of various lengths of Straight Pipe of Nominal Diameter from $\frac{1}{8}$ in. to 3 in.

Length of Pipe, Feet	Nominal Diameter of Pipe—Inches									
	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	3
10	3	14	33	72	130	240	—	—	—	—
20	—	7	18	49	88	165	340	530	—	—
30	—	—	12	38	70	130	270	420	890	—
40	—	—	—	29	60	110	235	360	770	2240
50	—	—	—	23	52	98	210	320	680	1980
60	—	—	—	—	47	89	190	290	620	1800
70	—	—	—	—	40	81	170	260	560	1630
80	—	—	—	—	—	75	160	240	520	1510
90	—	—	—	—	—	71	150	230	490	1420
100	—	—	—	—	—	—	140	215	460	1340
150	—	—	—	—	—	—	110	170	370	1070
200	—	—	—	—	—	—	96	150	310	990
250	—	—	—	—	—	—	—	130	280	795
300	—	—	—	—	—	—	—	115	250	710

For mains and larger diam. pipes use Pole's Formula—$Q = \sqrt[2]{\frac{pd}{sl}}$ where

Q = discharge cu. ft. of gas per hour,
d = diam. of pipe in inches,
p = pressure drop in pipe—inches water gauge,
s = specific gravity of gas (air = 1·0)—usually taken as 0·5,
l = length of main in yards.

Table LIVa gives the increase in resistance due to bends and tees.

TABLE LIVa

Resistance of Fittings

Nominal Diameter of Pipe	Addition to be made to Overall Length of Pipe in Feet for Increased Resistance due to Fittings		
Inches	Elbows	Tees	90° Bends
$\frac{1}{2}$ to 1 - - - -	2	2	1
$1\frac{1}{4}$ to $1\frac{1}{2}$ - - -	3	3	$1\frac{1}{2}$
2 - - - -	5	5	2
3 - - - -	8	8	3

The figure of ·5 for the specific gravity is usually accepted as a standard one, but should the figure be different in any particular case, the necessary

correction can be made by adding 1 per cent. to the flows for every ·01 below the standard gravity of ·5 or vice versa.

The underlining of certain figures in the Table indicates the maximum length of any particular diameter of pipe it is desirable to lay.

In normal gas supply practice the maximum pressure difference between the main and any gas apparatus supplied from it should not exceed five-tenths inches water gauge. This is usually allocated as one-tenth inch to the service, two-tenths inch to the meter and two-tenths inch to the piping (commonly termed the carcassing) in the building.

Gas being lighter than air, the pressure in a gas pipe rises the higher the gas pipe is carried. The amount of increase is given below:

Sp. Gr. of Gas.	Height for Rise of $\frac{1}{10}''$ W.G. Pressure
0·46	13 ft.
0·50	14 ft.
0·54	15 ft.

In normal practice the gas-consuming apparatus is higher than the gas main, and this will account for one or two tenths w.g. increase in pressure, increasing the allowance for friction in the pipes from two to three or four tenths. The flow figures given in Table LIV can then be used as follows:

Divide the gross length of the pipe (i.e. net length plus additions for bends, etc.) by the allowable pressure drop in tenths, and thus obtain the gross length corresponding to the standard drop of one tenth. By referring to the table, and knowing the gas volume required, the size of the pipe necessary may be found.

Gas pressure in the mains varies during the day, due to the incidence of the peak demand, and for large buildings a governor is usually fitted on the service to maintain a constant pressure on the apparatus. Most modern apparatus has, however, a gas governor incorporated in the design, and generally governors to ordinary houses are not necessary. In very high buildings the build-up of pressure in the internal piping may be very great, and in these cases governors are fitted on certain floors to break it down, so that the equipment will work satisfactorily.

Gas, as it leaves the works, generally contains water vapour, and as the temperature falls when the gas passes along the pipes in the ground, condensation may take place. Gas services to buildings should be laid, therefore, with a slight fall to a low point where the water can be collected in a syphon or collecting chamber. When the pipes enter the building, the temperature is usually higher, and condensation does not then occur, so there is no necessity for any gradient being made in the pipe. Plugs and tees should, however, be left in convenient positions to enable any corrosion products on the inside of the pipe or other deposits to be removed.

A number of gas undertakings are now supplying gas which has been partially dehydrated and washed to remove moisture and naphthalene, so that the necessity for syphons and gradients is then not so important.

CHAPTER XIII

Heating and Hot-Water Supply by Electricity

Electrical heating has the following good points, some of which it shares with gas:

(*a*) Absence of fumes or products of combustion.
(*b*) Avoidance of labour.
(*c*) Availability at any temperature at point of emission, hence great freedom in design of apparatus.
(*d*) Ease of thermostatic control.
(*e*) Transmissibility to any point regardless of levels or limitations found with other media.
(*f*) Shortness of time lag.

Two vital considerations, however, limit the free use of electricity for heating:

(1) Cost.
(2) Generating capacity.

(1) *Cost.* The cost of current is determined by the price of coal, the losses in generation, the interest and depreciation on plant and distribution system, plus administration cost, labour, etc.

The losses in generation are fundamental. Fig. 173 shows diagrammatically the transition which takes place from the heat in the raw coal, taken in by the generating station, to the current received by the consumer.

The first loss occurs in the boiler, due to the residual heat in the flue gases, radiation, etc., just as in an ordinary heating boiler, except that power station boilers commonly work at higher efficiencies of between 80 and 90 per cent.

The second, and by far the greatest loss, occurs in the turbine in converting heat energy into mechanical energy at the shaft of the generator. A study of thermodynamics will show that when working between the temperature limits at present possible in practice, the efficiency of conversion cannot be more than about 40 per cent., whilst with older stations it may be 25 to 30 per cent. The heat rejected comes out at a low temperature of about 90° to 100° F. in the condenser water, which is passed to a river, canal, lake or cooling tower.

A third loss occurs in the generator or alternator in conversion of mechanical energy into electrical energy.

A fourth small loss occurs in the units consumed in auxiliaries in the process of generation, such as in coal-handling plant, forced and induced draught fans, feed and circulating water pumps, etc.

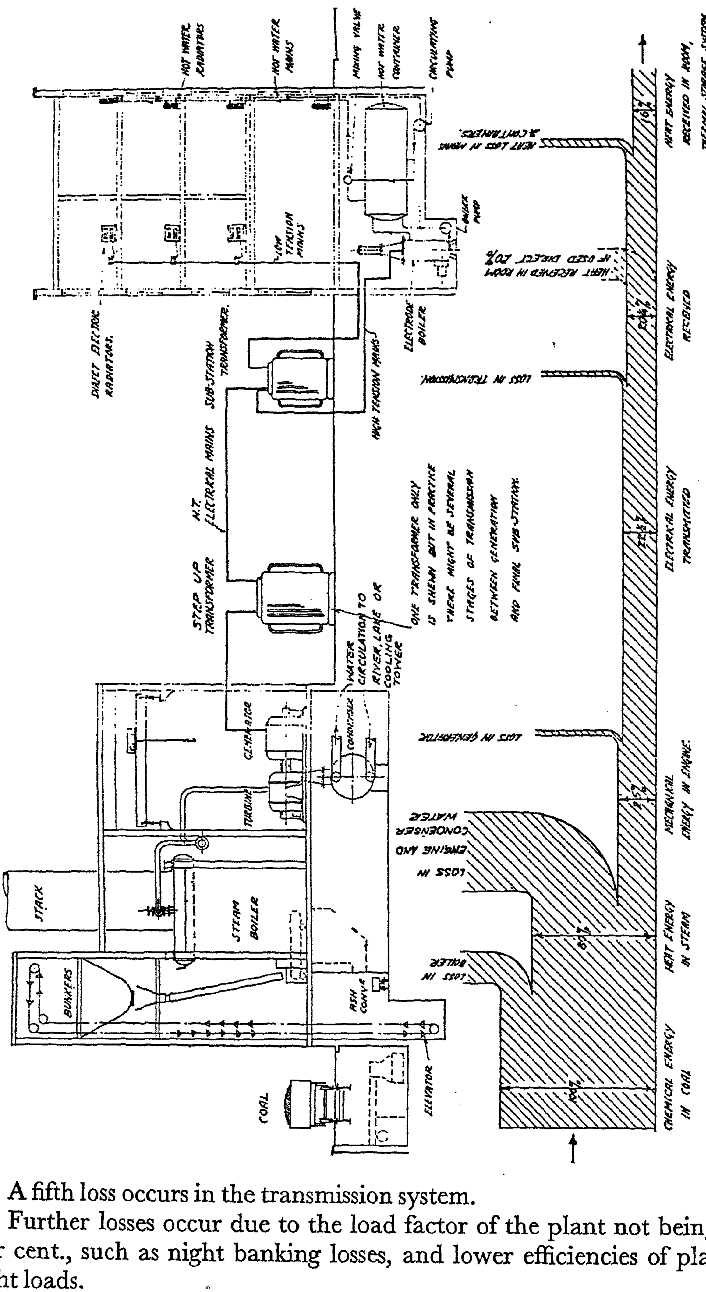


Fig. 173.—Diagram of Electrical Transmission of Heat.

A fifth loss occurs in the transmission system.

Further losses occur due to the load factor of the plant not being 100 per cent., such as night banking losses, and lower efficiencies of plant at light loads.

The final result is that the ratio of heat delivered to a building to heat

input of coal supplied to boiler at present lies between 20 and 25 per cent. in actual practice, but theoretically it can reach 30 per cent.

This may be seen in another way. The coal consumed to *generate* one unit of electricity expressed as an average over all the stations in Great Britain in 1938 was 1·4 lbs., and the lowest of any station was ·85 lbs.

Taking coal as fired at 12,500 B.T.U./lb. calorific value, and the heat equivalent of a unit of electricity as 3415 B.T.U., it will be apparent that the average efficiency of conversion was

$$\frac{\times 100}{1{\cdot}4 \times 12500} = 19{\cdot}6 \text{ per cent.}$$

From this has to be deducted the transmission losses which, if taken as 10 per cent. of the current generated, gives a net efficiency of 17·6 per cent. at the consumers' terminals.

On the above basis the average coal cost per unit, taking coal at 16s. per ton, will be

$$\frac{1{\cdot}4 \times 16 \times 12}{2240} = {\cdot}12\text{d. per unit generated.}$$

To this has to be added the other items of cost enumerated above, such as labour, transmission losses, sinking fund charges, maintenance, etc. These amounted in 1938 to ·26d. per unit, taking an average over the whole country. It will easily be seen that prices of less than ·38d. per unit are unremunerative at present. The coal consumption per unit has, however, been steadily falling and the output steadily rising for many years.

Current on demand at any time required for direct electric heating necessitates generating plant adequate for the peak load, which may be coincident with the peak load for power, lighting, etc., as on a foggy day in winter. Such load is not attractive to supply authorities, since it increases their maximum demand on the grid or on their station. For direct heating, rates are therefore commonly charged around ·5d. per unit upwards, often coupled with a standing charge, or maximum demand charge, bringing the actual cost to ·6d. or ·75d. or even more.

In order to remove this difficulty more advantageous rates are generally offered for current supplied off peak load. This means that heat is shut off by time switch, or otherwise, during specified hours in certain months. With direct electric heating this is a serious disadvantage, and means have been devised for taking current off peak, generally at night time, and storing it as heat for use during the day. This method is referred to later under 'Thermal Storage'. Rates offered by many Supply Companies for this type of load are around the basic minimum of ·2 to ·3d., since it involves no additional plant or labour. By the adoption of this system it is often possible to obtain lower rates for current used for other purposes, so that a direct comparison of costs cannot be covered by a generalization. The

particular balance of costs in any one case must be examined very carefully before a strict comparison can be made.

The thermal storage system, however, robs electricity of one of its advantages for heating, in that it involves the use of a heating medium such as hot water, with its pipes, radiators, and storage vessels, longer time lag, radiation losses, and more limited control and application as compared with direct electric heating.

The question of costs may be summarized as follows:

Normal heating rates - -	0·6d., 0·75d., or more per unit
equivalent to - - -	17·52d. to 21·9d. per therm

available at any time for direct heating.

Off peak load rates - - -	0·2d. to 0·3d. per unit
equivalent to - - -	5·84d. to 8·76d. per therm

possible (generally) only with thermal storage system.

Reference to Table XV, p. 51, will show that these rates at first sight compare unfavourably with, say, coal or coke (at 40s. per ton) equivalent to 1·8d. a therm. Some qualification is, however, necessary.

Direct Electric Heating is 100 per cent. efficient. Thermostatic room control limits heat input to exact requirements. Requires no labour.

Thermal Storage Heating. Conversion to heat 100 per cent. efficient. Radiation and mains losses, say, 15 per cent. Thermostatic control of water temperature does not limit heat input so exactly as with direct system. Wasted heat may account for about 10 per cent. loss. The radiation losses are continuous 24 hours a day, so that the above loss is increased for intermittent heating.
Small amount of labour only.

Solid Fuel-fired System.
Boiler efficiency 70-75 per cent. maximum, over a period probably about 65 per cent. efficiency.
Radiation and mains losses, say, 15 per cent.
Thermostatic control not so exact as direct room control; also residual losses at night, etc., may account for about 15 per cent.
Labour required for operation.

On the above arbitrary basis the overall efficiency of the systems comes out as follows:

Direct electric heating - - - - -	100 per cent.
Electric thermal storage heating, $100 \times \cdot 85 \times \cdot 9$	76·5 ,,
Solid fuel system, $65 \times \cdot 85 \times \cdot 85$ - - -	47·0 ,,

Price per therm is then:

Direct (·6d. per unit) - - - - -	17·52d. per therm
Electric thermal storage (at, say, ·25d. per unit)	9·56d. „ „
Solid fuel system (coal or coke at 40s. per ton = 1·8d/therm) - - - - - -	3·83d. „ „

The latter requires some adjustment for labour. Convenience and cleanliness may also have a value. The corresponding figures with gas and oil fuel can be similarly estimated.

The comparison with direct electric heating is also affected by the period of utilization, i.e. whether heat is required intermittently or continuously. For example, with continuous heating the advantage of electric heating becomes less. These matters, as explained, render generalization impossible, but some further reference to them is made under Chapter XVI on Running Costs.

Generating Capacity

The limitation previously referred to under this heading is due to the enormous loads of electric heating if carried out on a wide scale, as compared with other electrical loads such as lighting, power and cooking.

A glance at Fig. 174 will show the proportion of heating load to other loads as actually taken for a normal building. When considering a vast network of supply mains, say, for a large town, the matter is worse than this, since with lighting, power, cooking, etc., it is found that there is a diversity factor due to the fact that the maxima do not coincide. The factory power load, for instance, will probably not coincide with that for cooking, and so on. In the case of a residential district we get a load curve somewhat as shown in Fig. 175. But in the case of heating, in cold frosty weather, it is highly probable that all heating load will be on simultaneously with one or other of the different peaks.

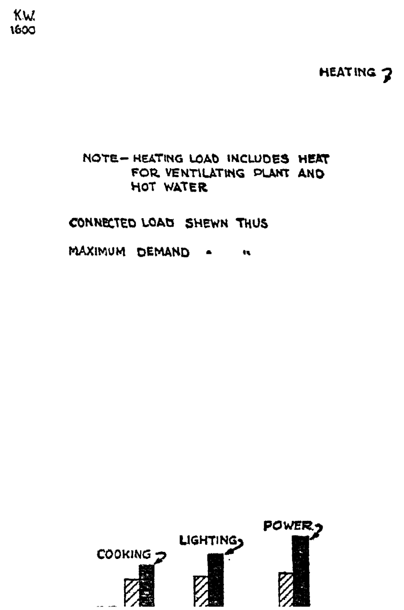


FIG. 174.—Diagram showing Relative Electrical Loads compared with that for Electric Heating. Taken for Actual Building of 1·43 Million Cubic Feet Gross Cube.

If all electric heating were by thermal storage with night-time supply of current, the other peaks would not be coincident, but even so the vast load would be greatly in excess of existing generating plant and mains.

So far thermal storage electric heating has been developed only in areas where capacity existed without serious addition, and particularly where the extra night load has filled up the valleys in the load curve so as to give an improved load factor and all-round rise in efficiency.

To increase the night load beyond the present-day maximum would obviously reverse the order, and low rates at night would no longer be possible.

At present the installed capacity of public generating plant in Great Britain is about 10 million kilowatts, having increased rapidly from about one-third of this figure twenty years ago. One million kilowatts represents

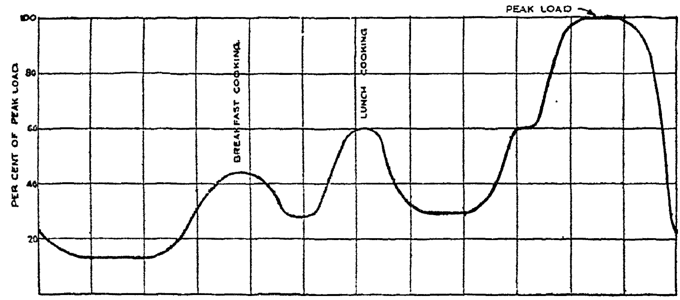


FIG. 175.—Diagram of Typical Daily Load Curve for Residential District.

3,415 million B.T.U.'s per hour. This is by a rough computation the load which would be required to heat all the buildings in the small area covered by the City of London, being about one-hundredth of the area of the County of London. As a matter of interest, it may be mentioned that the completed Battersea station has a generating capacity of about half a million kilowatts. Thus the total generating power in the country could heat an area ten times that of the City of London if it were all devoted to that purpose. This area is, of course, the most congested in the country.

It is obvious that the continued growth of electric heating is severely limited by the capacity of existing generating plant, transformer stations and distribution mains. To increase these solely for heating, required only in winter, may not be economic when other means exist for achieving the same end much more cheaply.

This aspect of the matter should be kept clearly in view when considering heating in connection with large-scale developments, though it is clear that electrical heating in one form or another will find an increasing application in a proportion of building schemes.

DIRECT ELECTRIC HEATING

If electricity for direct radiation costs ·6 to ·75 pence a unit (17·52 to 21·9d. per therm) as against 3·83d. per therm for coal, or coke with central heating, it is clear that for *basic heating load over the whole heating*

period, electricity is generally too expensive. But it is pre-eminently suitable for producing immediately and for short periods a beam of radiant heat, as may be required, for example, for dressing and undressing in an otherwise cold bedroom. Most healthy people dislike sleeping in a warm bedroom, and there is a general opinion that it is healthier to sleep in a cold bedroom with open windows, provided one is used to it. But the heat given by an electric fire with reflector is then very acceptable, and enables clothes to be quickly warmed in the beam before being put on.

If such a fire is left on for, say, 20 minutes in the morning and 20 minutes at night—say, 40 minutes a day—its cost is the same as that of a radiator heated by coal-fired boiler left on for 5 times as long, say, $3\frac{1}{2}$ hours ($1\frac{3}{4}$ hours morning and evening). The boiler and radiator would certainly need to be left on as long as this for any benefit to be derived from them because they have such a great time lag, whereas the electric fire is full on in a minute or so. Hence the electric fire will always find a sphere of great usefulness in our heating economy.

Load to be installed—In the following discussion it is assumed that, instead of the local hot-spot heating referred to previously (for dressing, etc.), it is desired to heat a whole room or building up to some specified temperature, such as 65° F. with 30° F. outside. The electrical loading in kilowatts of heaters to be installed in a room or building depends, as in the case of all other forms of heating, on the heat losses.

$$\text{Thus K.W. to be installed} = \frac{\text{Heat loss in B.T.U./hr.}}{3415}.$$

To this must be added a factor to cover building time lag, as explained in Chapter III. If, with continuous heating, the net load only is installed it will theoretically take an infinite time for the room to arrive at the designed condition. With intermittent heating a larger margin is necessary in order to shorten the lag. There is no possibility of forcing the system, and hence no hidden reserve to draw upon. Thus, in addition to a margin for time lag, it is necessary to consider a further addition to cover against the outside temperature being below the 30° F. of the calculations. This may, of course, equally well be allowed for by taking out the heat losses for, say, 25° F. outside.

The above considerations will often require an addition of between 50 and 100 per cent. to the basic figure, but whilst this is necessary with convective and low temperature radiation heaters, it appears that with high temperature luminous heaters a lesser margin suffices. This is no doubt due to the immediate comfort effect of visible heat radiation, more or less irrespective of air temperature. As mentioned in the discussion on gas heating, the design of such a system should be based not on normal heat losses, but on a calculation of mean radiant effect, coupled with the indirect heating of the air and re-radiation from walls, furniture, etc. Such a computation is very involved, and no simplified method appears to have been arrived at.

It is found that high temperature radiant systems, if based on normal heat loss calculations, meet most requirements if installed with 20 to 25 per cent. margin.

Building Insulation: Electric Heating—The more expensive the fuel, the more does it pay to insulate the building. This was shown to be the case in the section on Insulation in Chapter II.

With direct electric heating the building should be insulated to the utmost practicable extent. If considered at the outset this does not always mean greatly increased cost, as there is a substantial saving on cost of heating installation to offset against it, apart from the saving in running cost. If fullest use is made of building insulation, thereby cutting the heat losses down by perhaps a half or one third, direct electric heating may be found to be practicable where otherwise it would have to be abandoned as too expensive.

Direct Electric Heating Equipment—Various types of apparatus were illustrated, or referred to, in Chapter IV. Some idea of loadings were also given. In order of temperature of source they may be classified as follows:

Luminous radiators, portable and fixed	1800–2000° F. approx.
Non-luminous high temperature panels	500–600°
Electro-vapour radiators - - -	200°
Low temperature panels - - -	160°
Tubular heaters - - - -	150°
Convectors - - - - -	130° (air temp.)
Unit heaters - - - - -	120° (air temp.)
Low temperature panels, embedded type	100°

New developments include radiators in which oil is used as the medium; panels in which the element is embedded in a plastic material; and soap-stone radiators which are in effect small thermal storage units by which it is possible for current to be switched off at peak load. With the latter the lower rates previously mentioned for thermal storage systems would often apply.

There are, in addition, various combinations of convectors with luminous heaters, convectors with fans, panels with convection passages behind, etc. It is not necessary, or possible, to deal with the applications of all types here, especially as they continue to multiply, but reference will be made to one or two typical cases.

Case 1 is a works canteen, as shewn in Fig. 175 (*a*). Size 84 ft. × 26 ft. × 10 ft., cube, 21,800 cu. ft., heat loss 65,400 B.T.U.'s/hr., seating 200 occupants, natural ventilation by roof vents. This is satisfactorily heated by 10–2½ K.W. luminous parabolic trough type radiators fixed overhead, some 8 ft. above floor. They are switched on and off by hand, as they are not suitable for thermostatic control. This would then cost about 1s. an hour to heat (at ½d. a unit), or less if only partly on.

This kind of heating, as mentioned above, has a quick response, and is particularly suited to intermittent use, as in the case above, and for small assembly halls, churches, institutes, restaurants, workshops, etc., etc.

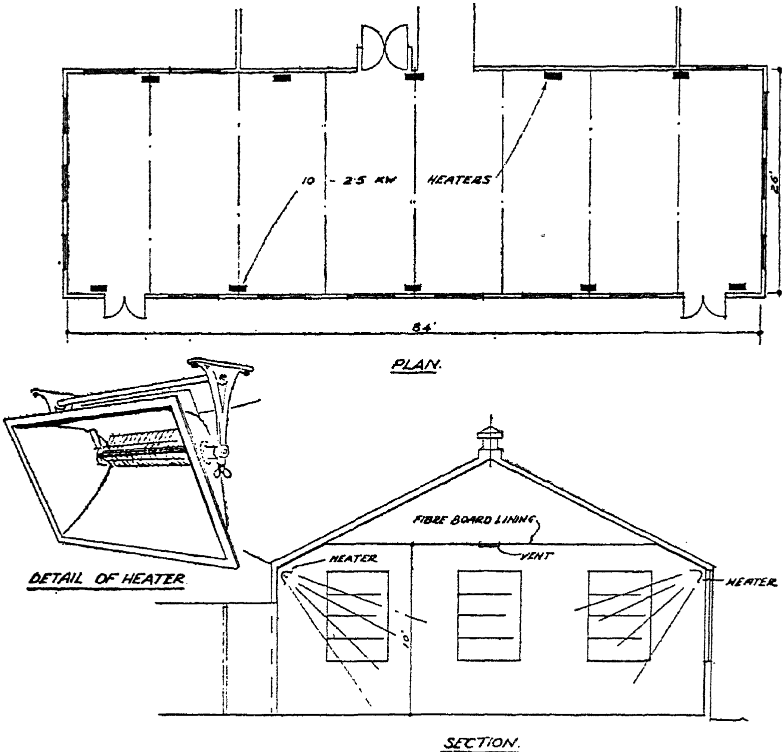


FIG. 175 (*a*).—Application of Luminous Electric Heaters to Small Canteens.

One disadvantage of this type, if fixed near to a white ceiling, is that the vertical convection currents cause serious blackening immediately over each heater, especially in dirty atmospheres.

Another disadvantage, in very cold weather, is that the feet are inadequately warmed, being shielded from the heat radiation.

Case 2 is of a normal office building, in which the rooms are heated by electric tubular heating, or might be by any type of convection heating as shown in Fig. 175 (*b*). The heaters are switched on and off by thermostat placed on the wall. The installed load has a margin of 50 per cent. over net heat losses.

Case 3 is of a small workshop in which electric unit heaters are installed, of the type shown in Fig. 175 (*c*), each controlled by thermostat set to 60°. Electric unit heaters are particularly useful for space heating in isolated buildings, and similar cases where a central system cannot be connected.

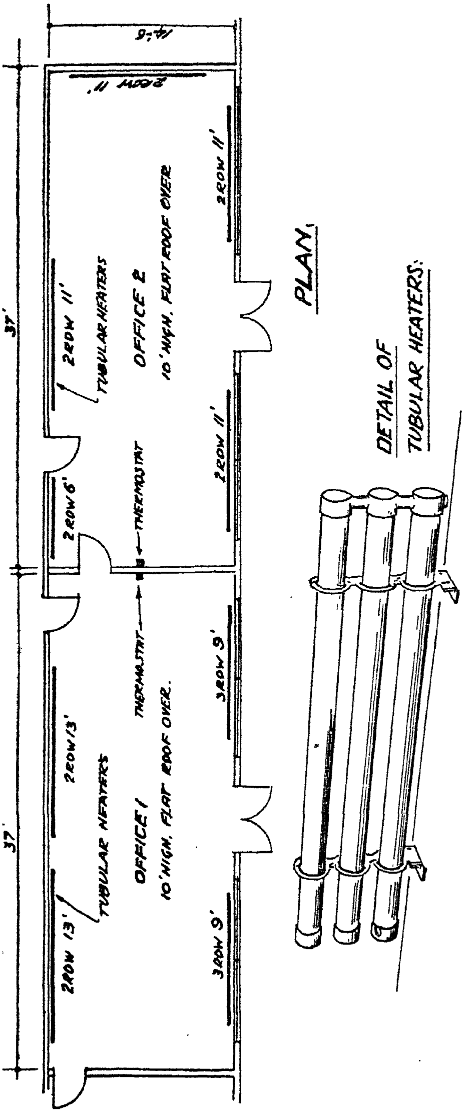


FIG. 175 (*b*).—Electric Tubular Heating to Offices.

Each is thermostatically controlled. These units are made in sizes 5, 10, 15
d

Case 4, shown in Fig. 175 (*d*), is a school, heated by inclined radiant heaters of non-luminous type. This form of heating is more economical for buildings where the ventilation rate is high, than the convective type of system. The air is warmed indirectly, the main heating being by radiation,

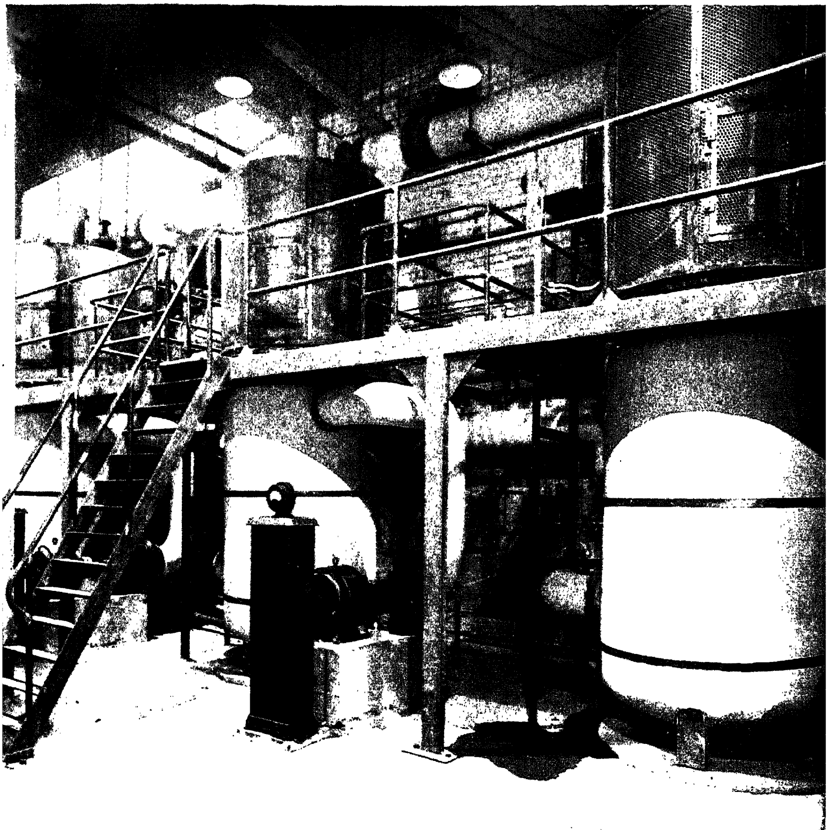


Plate XVII. Electric thermal storage system at Earl's Court, London, showing three Electrode boilers, 14,000 KVA total. Installed by Bastian and Allen, Ltd. for the Fulham Borough Council
(see p. 292)

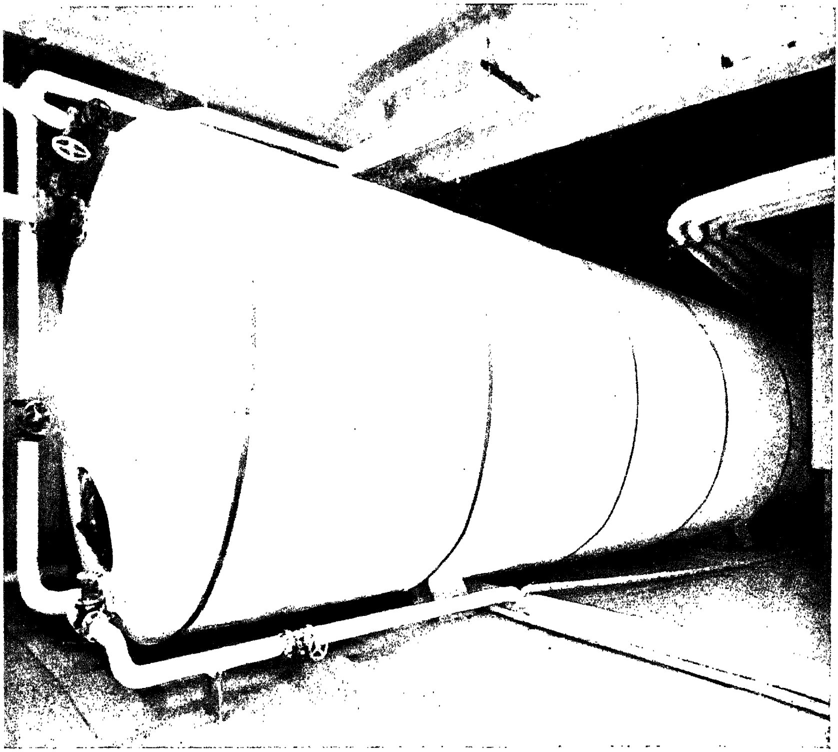
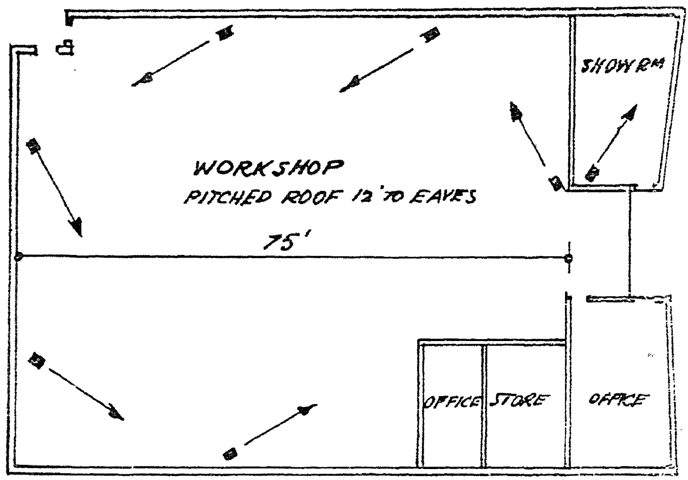


Plate XVIII. Thermal storage cylinder, one of seven at Earl's Court. Total capacity 170,000 gallons *(see p. 292)*



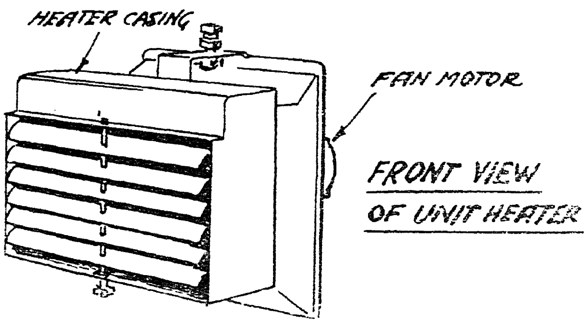


FIG. 175 (*c*).—Workshop heated by Electric Unit Heaters. (G.E.C.)

and for this reason control is better performed by eupatheostat (see p. 74) rather than by plain thermostat.

Thermostats, for controlling the various types of electric heating system, are either bimetallic or vapour expansion type. The contacts may be open to the atmosphere, enclosed in a sealed tube, or by tilting mercury in a sealed tube. The load which it is possible for a thermostat to carry by direct switching varies with different types, but most will handle up to 15 amperes on alternating current (equivalent to 3 K.W. at 200 volts). On direct current the rating is reduced to about one-tenth.

For currents in excess of 15 amperes on alternating current and 1 to 2 amperes on direct current a *contactor* is generally employed. This consists of a single or double pole switch operated by electro magnet. The thermo-

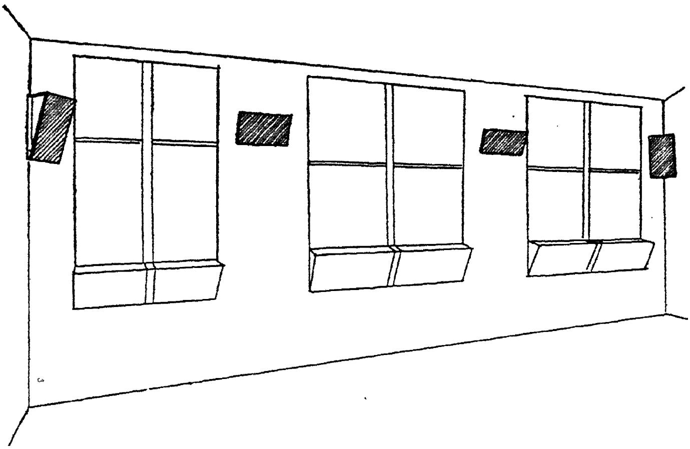


FIG. 175 (*d*).—Schoolroom heated by Electric Non-Luminous Radiant Panels. (Morganite.)

stat then makes and breaks the coil circuit only, which carries a current probably of about 0·1 ampere.

INDIRECT ELECTRIC HEATING

Electricity used for heating water or generating steam for heating purposes indirectly without thermal storage possesses none of the advantages of direct heating, and does not make possible use of 'off-peak' current. Its application is so limited that it need not be discussed here.

ELECTRIC THERMAL STORAGE

The case for this system will have been appreciated from the discussion earlier in this chapter, where it was stated that electricity 'off-peak load' can often be bought at rates as low as ·2d. a unit or 5·84d. a therm. It should further be noted that no flue or labour is required, and a comparison of costs should take this into account, together with its cleanliness and absence of smoke or ashes. Plates XVII and XVIII (pp. 290–1) show the 14,000 k.V.A. plant at Earl's Court, where the Authors were responsible for the design of the heating plant.

The system is indirect. The heat is stored in the form of hot water raised to as high a temperature as possible, and contained in large cylinders generally placed in the basement. The heat is generated either in an electrode heater utilizing the resistance of the water as the element, or in smaller systems by immersion heaters. The working of the apparatus is automatically controlled by thermostatic and time switches.

Operation—Fig. 176 shows the general arrangement of a typical thermal storage system. The principle of operation may be described as follows:

(*a*) The electrode water heater (or separate immersion heater) warms the water in the storage cylinder, the pump accelerating the circulation. If the immersion heaters are contained in the main storage no pumping is necessary.

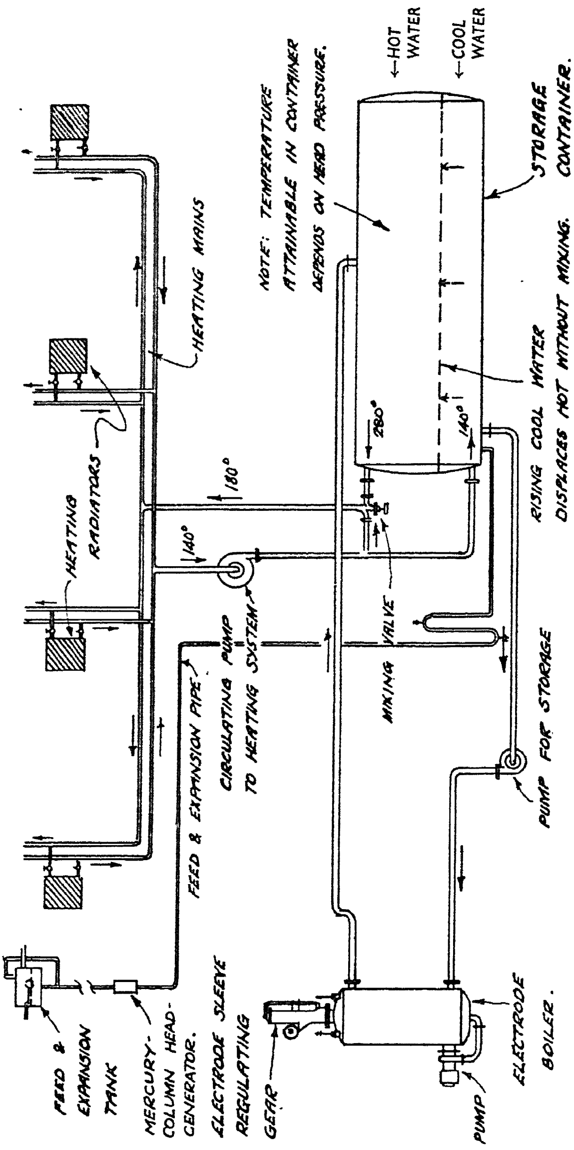


Fig. 176.—Diagram of Electrical Thermal Storage System.

(*b*) This heating takes place at off-peak hours, and in order to economize in capacity of storage the temperature is raised as high as possible without generating steam. This may be as high as 300° F., and is determined solely by the height of the building. An artificial head can be produced with a mercury

seal or 'heat generator' in the feed and expansion pipe, but this is not usual in this country. The storage temperature is generally kept about 20° F. below that at which steam would be produced.

(*c*) At a given time the current to the heater is cut off automatically by time switch, or from the supply company's sub-station. At the same time the storage pump is stopped.

(*d*) When heating in the building is called for the heating pump is started, high-temperature water is drawn off from the cylinder and returned cold at the bottom. A proportion of return water is mixed with the flow through a thermostatically controlled mixing valve. Thus, with possibly 280° in the cylinder, 180° may suffice for the radiators. If the system has gravity circulation and no pump, the operation is controlled by an electrically operated valve.

(*e*) This process continues all day, the level of cool water at the bottom of the cylinder gradually rising, but not mixing with the hot, and by night time in cold weather probably little hot water is left in reserve.

(*f*) When heating is no longer required the secondary pump is stopped.

(*g*) At a given time, when off-peak current is available, the supply is turned on to the boiler and the process repeated. The primary pump is simultaneously started.

(*h*) If the cylinder is completely emptied of high-temperature water during the day it is possible to have what is called a 'day boost'. That is to say, power is turned on to the heater during on-peak hours. For this, however, a higher rate is charged, and it is only resorted to in an emergency. Thus the storage capacity is controlled by the heat output of the plant, its period of use and the temperature to which it can be run. The electrical loading of the heaters is likewise determined by the daily total heat load and the number of hours during which current is available.

Types of Heater—The heater may be of two types according to the voltage and size of the installation:

(*a*) *Immersion heaters*, consisting of resistance elements inside metallic tubes or blades. These are suitable for low voltages up to 250, or when in a balanced arrangement up to 500, and are generally of 3 to 4 kw. loading each, arranged in groups or banks of 50 kw. or more. This type of heater is often placed direct in the bottom of the storage vessel as in Fig. 177. Installations rarely exceed 300 kw. Fig. 178 gives an illustration of a typical plant.

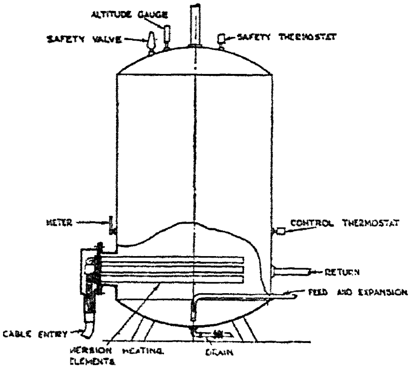


FIG. 177.—Immersion Type Thermal Storage Heater.

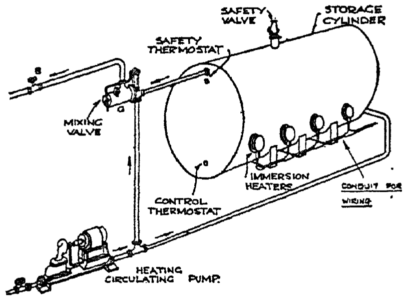


FIG. 178.—Diagrammatic Lay-Out of Electric Thermal Storage Plant with Immersion Heaters fitted direct to Storage Cylinder.

(*b*) *Electrode heaters*, connected to a medium voltage (400-600 volts) or high voltage (600 to 11,000 volts) three-phase alternating current supply. They are suitable for installations up to 5000

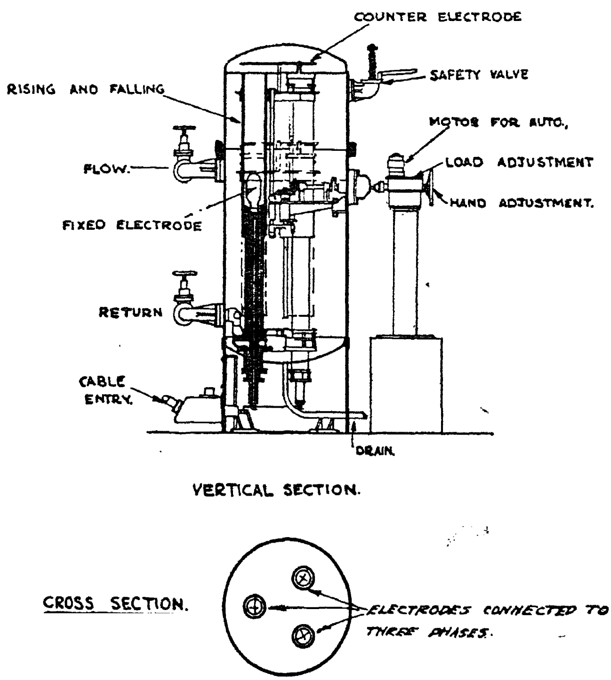


FIG. 179.—High Voltage Electrode Water Heater (Reyrolle).

kw. each or more, and one type is shown in Fig. 179. They are vertical (with one exception) and the chief difference in the various designs is in the method adopted for load regulation. Current passes from electrode to electrode, using the resistance of the water itself as the heating element, and the load is varied by increasing or decreasing the length of path which the current has to take by the interposition of

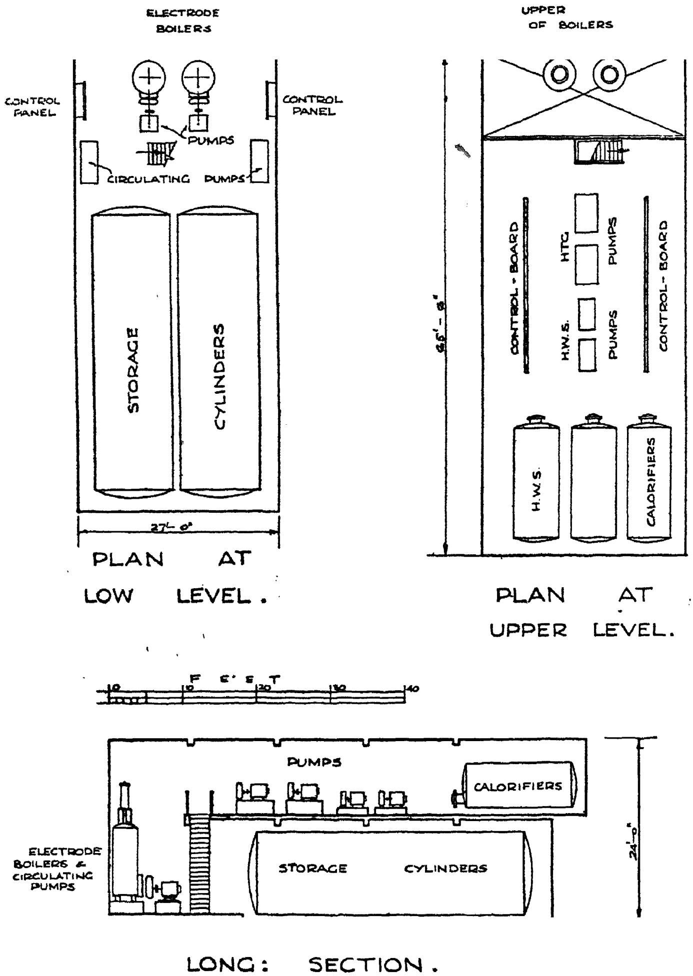


FIG. 180.—Lay-Out of High-Tension Thermal Storage Plant. Total Capacity, 3200 kw.

non-conducting shields between or around the electrodes. The conductivity of the water has to be adjusted by the addition of soda, or other salts if need be, to render it suitable

for the purpose. This type of heater is kept separate from the main storage because means of load regulation involving raising and lowering or rotating gear to the sheaths complicates a direct application; and, further, one heater may be connected to a series of storage vessels to provide the capacity required. Renewal of the electrodes from time to time is also simplified where they are in a separate vessel.

Fig. 180 shows plan of typical high voltage plant.

Loading of Heaters—Thermal storage electrical load in kw. per twenty-four hours

$$=\frac{\text{Total B.T.U.'s required per day}}{3415}.$$

The total heat requirements must include all radiation and other losses over the period.

The hourly load in kw. depends on the duration of the off-peak supply, which may be anything from eight to eighteen hours. Hourly kw. of plant

$$=\frac{\text{Load per twenty-four hours}}{\text{Duration of supply in hours}}.$$

Thermal Storage Cylinders—Storage vessels are most conveniently and cheaply made cylindrical and preferably horizontal, the size and shape being largely determined by space conditions. Diameters up to 10 ft. 6 in. are usual, with lengths up to 30 ft.

The storage capacity to be provided depends on the total heat requirements per day and on the temperature range.

Capacity in lbs. of water

$$=\frac{\text{Total heat required during time current is off}}{\text{Storage temperature} - \text{Minimum heating return temperature}}.$$

The capacity in gallons may be arrived at by dividing the lbs. by the weight per gallon at the storage temperature, as determined from the following table:

TABLE LV

Temperature °F.	Weight per Gallon, Lbs.	Head Required on Cylinder, Ft.*
200	9·62	—
220	9·56	25
240	9·47	50
260	9·38	80
280	9·28	120
300	9·18	175

* These are on basis of boiling points 20° F. in excess of the temperatures given.

The dimensions of cylinders of various capacities may be obtained from Table XLVI (p. 233) or from Table LVI.

TABLE LVI

Diameter of Cylinder	Galls. per Foot of Length
4′ 0″	79
4′ 6″	99
5′ 0″	122
5′ 6″	148
6′ 0″	176
6′ 6″	207
7′ 0″	240
7′ 6″	278
8′ 0″	314
8′ 6″	348
9′ 0″	398
9′ 6″	443
10′ 0″	490
10′ 6″	540

Expansion of the heating water is much larger on thermal storage systems than with heating boilers, on account of the greater volume and the high temperature of the water stored. For instance, when water is raised from 120° to 240° F. the increase in volume is about 4¼ per cent., and to 300°, 7½ per cent.

If the water of expansion is allowed to pass heated from the water heater, and then is allowed to cool off in the expansion pipe and tank before returning gradually during the day, a certain loss of heat is incurred. For this reason a sufficient space at the bottom of the cylinders should be arranged below the level of the return connection to contain the diurnal water of expansion. From the bottom of this space the feed and expansion pipe is connected as in Fig. 181. When the water is heated it depresses that at the bottom up into the tank and in theory only cold water should be forced up the expansion pipe. In practice, there is probably some mixing, in addition to the small heat transmission by conduction, and a small loss from this source is unavoidable.

In any case, this expansion space is only considered to accommodate the daily expansion, i.e. from about 120° to the storage temperature, since the expansion from 50° to 120° only occurs once during the heating season and it is unnecessary to provide for it in the cylinder.

The height of the return pipe from the bottom of a horizontal cylinder should therefore be about one-eighth of the diameter, corresponding to an expansion allowance of 8 per cent.

Insulation—Loss of heat from the cylinders and heater is reduced as far as possible by efficient insulation. Three-inch cork is frequently used for this purpose, having an efficiency of about 93 per cent. Higher efficiencies are possible with 3 in. or 4 in. of glass silk, the latter having an efficiency

of about 97 per cent. The economic limit must be assessed before the best thickness is determined, but electricity being an expensive fuel a high efficiency of insulation is justified. Cradles are also insulated with hard

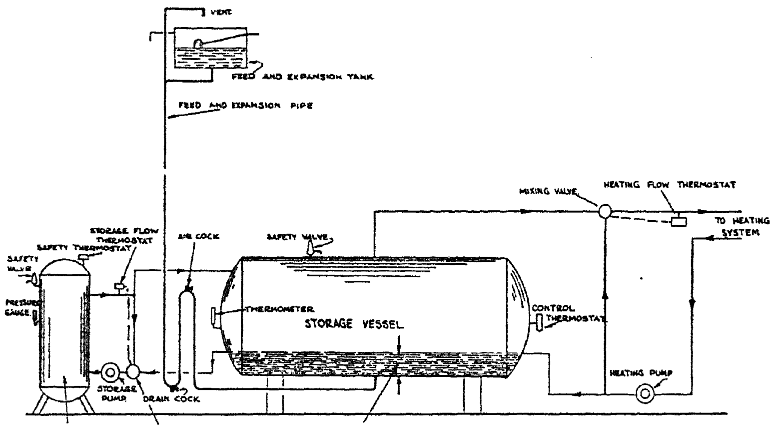


FIG. 181.- -Diagram showing Connection of Expansion Pipe and Mixing Arrangements of Electrode Heater.

material, such as compressed cork or hard wood, to minimize heat loss by conduction.

Mixing Valve—The mixing valve, which is an essential part of every thermal storage plant, consists of three ports. One is the high-temperature water inlet, one the cool return water inlet, and one the mixed water outlet. The proportions of the two former are controlled by a valve or valves operated by means of water pressure, electrical solenoids, or motor, from a thermostat in the mixed outlet pipe.

Expansion Tanks—As has been said, relatively large expansion tanks are required for thermal storage systems capable of containing the water of expansion resulting from

(*a*) heater and storage capacity raised from 50° to maximum storage temperature;
(*b*) heating system capacity raised from 50° to maximum working temperature.

Control—As the temperature in an electrode heater rises the resistance of the water becomes less and the load correspondingly goes up. Thus a 100 kw. heater at 100° F. would at 300° F. have an output of about 220 kw. Means to provide a constant outlet temperature, and hence constant load, are included by some installers, making use of a further thermostatically controlled mixing valve in the boiler-cylinder circulation as
181.

Protective gear is necessary in the case of electrode heaters to prevent operation on two out of the three phases, or with out-of-balance currents. Such faults might cause heavy earth leakage currents since the latter are only avoided when the current from all three phases is equal.

The protective and control gear for thermal storage systems is perhaps outside the scope of the present book, and each make of plant employs different methods for achieving the same purpose. It is a great advantage to have all the control instruments, relays, contactors and switchgear mounted on a common switchboard so that faults can be more easily located and proper supervision given.

HOT-WATER SUPPLY BY ELECTRICITY

As in the case of gas, hot-water supply may be provided electrically by local heaters or by a central plant, and the same advantages and disadvantages apply with either type, as was previously discussed.

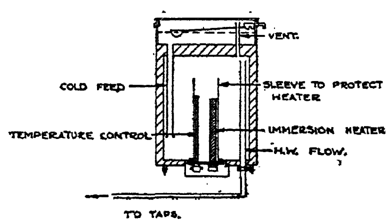


FIG. 182.—Open Tank Electric Hot-Water Heater.

Local Hot-Water Supply Heaters—Instantaneous heaters are out of the question owing to the heavy loads required, and storage type of low electrical input is the only one that need be considered.

These store the water at about 190° to 195° F., are thermostatically controlled, and are a very convenient method for domestic use where low rates for current are available, particularly in summer, when a boiler kept alight in the kitchen is a disadvantage. They are also very suitable for isolated basins and baths in buildings where a centralized supply is not necessary, and in such case may give greater economy than is possible with the latter.

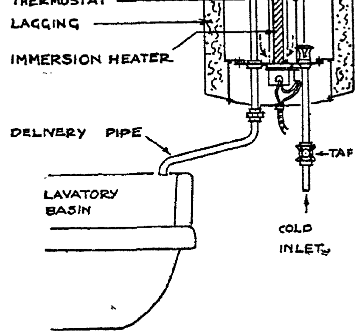


FIG. 183.—Electric Hot-Water Storage Heater; Closed, Non-Pressure Type.

Three types are available, as shown in Figs. 182, 183 and 184. Fig. 182 has a small open feed tank with ball-cock at the top and the cylinder is not under pressure. It is only suitable for taps at a level below the heater. Fig. 183 is closed, but again of non-pressure type, water being passed through when the inlet cock is opened. It is suitable only for a single point supply. Fig. 184 is a pressure

type which may be treated as a cylinder in a boiler system, pipes being run from it to the various taps, and the supply is derived from a feed tank in the roof.

Loadings usually range from 500 to 3000 watts, and capacities from $1\frac{1}{2}$ to about 30 gallons for domestic use. Larger units may be needed for other cases.

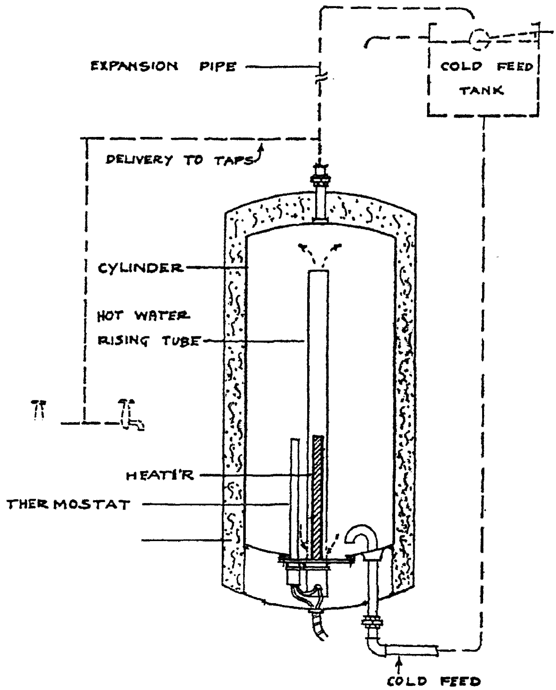


FIG. 184.—Electric Hot-Water Storage Heater, Pressure Type.

Sizes of less than 12 gallons are not adequate where a bath is to be supplied. Twelve gallons is stated to be adequate for one bath, 20 gallons for two baths in succession, 30 gallons for two baths plus other normal demands for kitchen, washing, etc.

Heaters of the above types, all of necessity efficiently insulated, incorporate thermostatic control and require no attention other than periodic descaling of the elements where the water is hard. Any circulating pipes from such heaters should be avoided as the consumption is increased considerably thereby. Towel airers and linen cupboards are best warmed by direct electric heaters.

Central Hot-Water Supply Systems—Where heating is accomplished by thermal storage in a large building hot-water supply requires similarly to be provided electrically from a central point.

Two methods are possible:

(*a*) By separate storage vessels heated by immersion heaters during the daytime, in which case a higher charge for current has to be faced.

(*b*) By using the high-temperature water from the thermal storage system through calorifiers as shown in Fig. 185 just as water from an ordinary heating system is sometimes used for providing hot-water supply on the 'indirect' method.

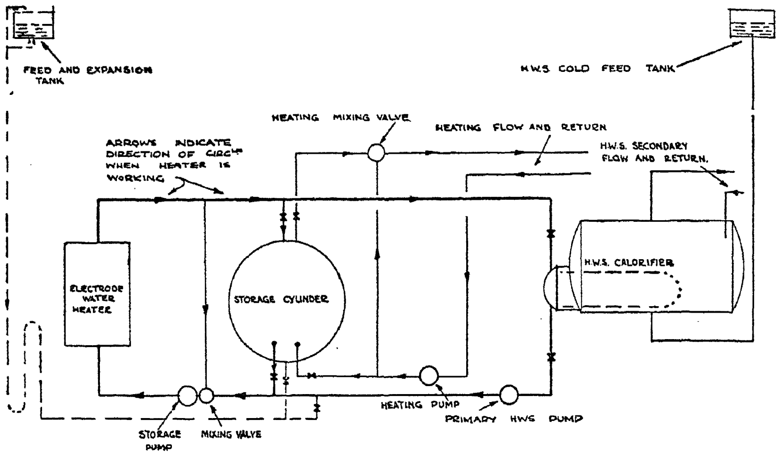


FIG. 185.—Diagram of Connection of Hot-Water System from Thermal Storage Plant.

The disadvantage of the latter method is that in the summer, when no heating is required, the heat losses of the large storage have to be maintained for what is, perhaps, quite a small hot-water load. Thus connections are sometimes so arranged that the electrode boiler may be used direct in circuit with the calorifiers, this again being with 'on-peak' current, but during summer only.

Whichever method is adopted, valves should be provided on the secondary circulations to be shut off at night, so as to save part of the heat loss due to radiation. In certain types of buildings, such as hospitals and hotels, where hot water may be required all night, this economy cannot be effected.

INITIAL COST OF ELECTRIC SYSTEMS

On the question of initial cost of electric systems, it may be generally assumed that direct heating by electricity is about the same in cost of installation as an ordinary hot-water system when wiring, etc., is taken into account. There may, however, be some saving in builders' work on the electrical scheme.

A thermal storage system is invariably higher in cost than a solid fuel-fired boiler system, the disparity being greatest on the smaller installations. A system costing £500 with coke will probably cost about £1000 with electrical thermal storage. If, however, the system is larger, costing, say, £5000 with coke, the thermal storage would probably increase it to about £8000.

These figures must be treated as a general indication only, as so much depends on the hours of off-peak current and other factors.

Hot-water supply from local electric heaters will be more economical than a central system for small systems where one, two or three heaters suffice. Above this, the first cost of a central system will generally be cheaper.

ELECTRICAL NOTES AND CALCULATIONS

Units—Current is measured in *amperes*. Pressure, or potential difference, in *volts*. Resistance in *ohms*. The current I passed through any resistance R when the potential difference is E, is given by Ohm's law:

$$I=\frac{E}{R},$$

so that

$$E=IR \quad \text{and} \quad R=\frac{E}{I}.$$

Power supply P is measured in *watts*.

$$1 \text{ watt}=1 \text{ volt} \times 1 \text{ amp.}$$

or $P=EI$. Thus power P, which is equivalent to the heating effect, is given by substitution:

$$P=I^2R.$$

It should be noted that a watt is a *rate* of doing work, and the unit of electricity is one kilowatt (i.e. 1000 watts) maintained for one hour (see p. 16).

This work in kilowatt-hours or units may be converted to B.T.U.'s at the rate of

$$1 \text{ kwh.}=3415 \text{ B.T.U.'s.}$$

Thus, to supply 1,000,000 B.T.U.'s per hour an input is required of

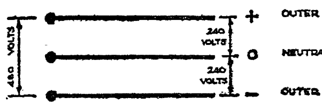
$$\frac{1{,}000{,}000}{3415}=292\cdot 8 \text{ kw.}$$

Types of Supply—Electricity is supplied direct and alternating.

With *direct current* one pole is maintained constantly at positive and the other at negative potential, so that the simplest supply is two-wire, one positive and one negative.

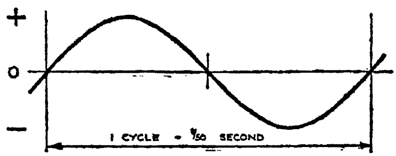
In a three-wire D.C. system a neutral or middle wire is introduced at earth potential, one 'outer' being maintained positive to this and the other

negative. The difference of pressure between the neutral and each outer is half that between the outers. Thus with a 480-volt D.C. three-wire supply the connection to the mid-wire would give 240 volts, thus:



When a three-wire D.C. system is 'balanced', the current taken by the positive-to-neutral exactly equals that from neutral-to-negative, and no current flows in the neutral. As soon as the load on either side is altered with respect to the other, the system becomes out of balance and current flows in the neutral either in one direction or the other.

With *alternating current* the polarity of the supply is reversed in regular cycles. Thus in a 50-cycle supply (standard in this country) the reversal takes place 50 times per second, and is said to be 50 cycle (~) current. The cycle is of sine wave form, thus:



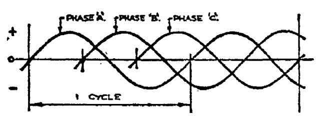
A.C. is usually either single-phase, two-phase or three-phase.

Single-phase current may be treated as D.C. supply for the purpose of heating loads which are all non-inductive and for which Ohm's law still holds good. The question of '*power-factor*'* does not enter on this account, i.e. power-factor is taken as 'unity'.

Two-phase three-wire supply is uncommon. Here the voltage between each of the outers and neutral is that between the outers divided by $\sqrt{2}$.

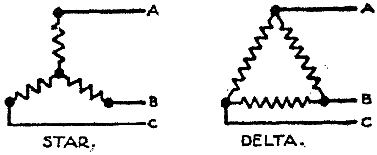
Two-phase four-wire supply, also uncommon, is virtually two single-phase supplies having the waves 180° out of phase. Each circuit passes half the current.

Three-phase three-wire may be visualized as three single-phase supplies 120° out of phase, represented thus:



* Where the load is inductive, as in a motor, the voltage and current get out of step with one another. The apparent watts as shewn by a voltmeter and ammeter are greater than the true watts as shown by a watt meter. The power factor is the ratio $\frac{\text{true watts}}{\text{apparent watts}}$.

The three wires may be connected to apparatus in star or delta formation:



The voltage between points A and B, B and C, C and A is the declared voltage, 400 volts is now standardized, but many others are in use.

Three-phase four-wire is the same as the three-wire, but a neutral line is introduced at the centre of the star formation, and this is earthed, thus:

The voltage between A and N, B and N, C and N, is then the declared three-phase voltage divided by $\sqrt{3}$. With the standard voltage of 400, it is $\frac{400}{\sqrt{3}} = 230$ volts.

Thus this supply is often referred to as 400/230 v. 50-cycle.

Each of the phases A, B, and C then give a single-phase supply, and a two-wire system is possible for each phase, using the neutral for the return of each.

When a three-phase four-wire system is balanced the load on each phase is the same and no current flows in the neutral. If any one of the loads is varied, current flows in the neutral wire.

This system is by far the most common and will become increasingly so.

Application to Heating Systems—For direct electric heating a two-wire system is essential. D.C. or A.C. single-phase at pressures under 250 volts are therefore alone used. As stated above, the D.C. may be obtained at this voltage from a three-wire system, and the A.C. from single-phase or a two- or three-phase system with a neutral return.

For thermal storage with immersion heaters any of the above low voltage supplies may be used, but the usual arrangement is with them balanced on a three-phase three- or four-wire supply.

Thermal storage with electrode heaters is invariably served from three-phase three-wire supply at medium or high voltage, i.e. 400 to 11,000 volts. A common voltage is 6600.

The current passing in the conductors may be calculated from the load and voltage, taking as an example the case above of 1,000,000 B.T.U.'s requiring 292·8 kw. (292,800 watts), the current passing through the circuit, will be

Two-wire D.C. *or single-phase* A.C. at, say, 200 volts

$$\frac{292{,}800}{200} = 1464 \text{ amperes.}$$

Three-wire D.C. at 400 volts across the outers, load balanced

$$\frac{292{,}800}{400}=732 \text{ amperes.}$$

(No current will pass in the neutral.)

Three-wire A.C., *two-phase*, at 400 volts between phase and neutral, load balanced

$$\text{current in phase line}=\frac{292{,}800}{2\times 400}=366 \text{ amperes per phase.}$$

The current in the neutral will be $366\times\sqrt{2}=518$ amperes.

Three-wire A.C., *three-phase* at 400 volts between phases, load balanced

$$\frac{292{,}800}{\sqrt{3}\times 400}=423 \text{ amperes per phase.}$$

Four-wire A.C. *three-phase* at 400 volts between phases (230 volts phases to neutral), load balanced.

The current will be the same as with the three-wire system, no current flowing in the neutral.

In the case of a number of small heaters, each will be served from a separate way on the distribution boards. Having established the current flowing in the circuit supplying one heater, these are totalled up to give the current supplied by the main to the distribution board. From these currents the electrical distribution system is then designed, but this is beyond the scope of the present book. For a more complete study, reference should be made to one of the many text-books on Electrical Engineering.

kW. and k.V.A.—The difference between kilowatts (kW.) and kilo-volt-amperes (k.V.A.) should be understood. The latter term is used in connection with alternating currents and refers to 1000 volt-amperes, or 1000 'apparent' watts. The kilowatt refers to 1000 'true' watts. At unity power-factor (see p. 304) the k.V.A. and the kW. are the same, but if the load is inductive then k.V.A. × power factor = kW. Normal heating elements are non-inductive and the terms are then synonymous, but such is not the case where motors or power transmission lines are concerned.

CHAPTER XIV

Heating by Steam*

There is, in this country, in buildings of the domestic type, such as residences, flats and offices, probably less than one steam-heating system for a hundred hot water. In the United States of America and Canada the reverse is the case.

The reason is not difficult to find. Our weather in winter is often mild, and only rarely do spells of three or four weeks of frost occur when full heating is required. Their winters are generally severe with months of snow and temperatures well below zero.

For our climate the hot-water system is most suitable, because its temperature can be varied so simply and over such a wide range. For consistently cold climates the steam system is satisfactory, and often much more economical in first cost, though the Spring and Fall periods call for some degree of temperature adjustment, which accounts for the development of systems such as the sub-atmospheric described later.

A disadvantage of any steam system when installed in small or medium-sized buildings is that the boiler requires some intelligent supervision. The water level of a steam boiler has to be kept steady within fairly narrow limits and, however many automatic devices are furnished to ensure this, they are liable to fail unless regularly overhauled. Too high a level will often cause water to fill the radiators, which as a result go cold. Too low a level may cause fracture of the sections, which is both costly and dangerous. If the solution of salts in the water becomes concentrated as a result of long use without proper flushing out, foaming or priming may result, again with the result of water being carried over with the steam into the radiators and causing erratic operation.

Steam traps, air release valves and the other more or less complicated accessories of steam systems in general all have their troubles and call for maintenance from time to time.

A further point to remember is that steam of any kind is unsuitable for use in the embedded panel systems, and whilst it has been used in the metallic plate and Rayrad type the surface is too hot for comfort. Ordinary radiators and convectors are the most suitable heating surface. The former need guards or grilles for protection against burning where children or hospital patients are liable to touch them. Convectors are more satisfactory and neater, though perhaps more difficult to clean. At any rate, convectors are displacing the ordinary radiator in the United States, just as in this country the various radiant systems are superseding it for the better work.

* Those unfamiliar with the properties of steam may first wish to refer to the Appendix at the end of this chapter.

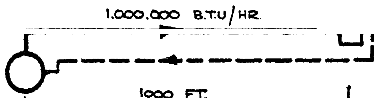
In passing, it is interesting to note that the tendency in the two countries is in opposite directions. The one (in America) towards smaller units using the principle of convection, and the other (here) towards larger and more extended surfaces relying on the principle of radiation. The causes leading up to this divergence include the following:

(*a*) Roughly $2\frac{1}{2}$ times the amount of heat has to be supplied in cold countries (with outside temperatures of 0° F. or below) as is required here. With low-temperature surfaces the large areas necessary would often be difficult to accommodate.

(*b*) The height of buildings common in the U.S.A. gives steam an advantage over water, in that with the latter the pressures on lower floors would be immense.

One characteristic of all steam systems is the rapid response or short time lag. This is due to the low heat storage of the system, depending on the high velocity with which steam travels, and to its high heat content per unit of weight, due to its latent heat.

One pound of steam at 1 lb. per sq. inch initial pressure, temperature 215° F., occupies 25 cu. ft., and carries a latent heat of 968 B.T.U.'s per lb. For conveying, for example, 1,000,000 B.T.U.'s per hour from point *X* to point *Y* in the sketch, 1000 ft. apart, with water and with steam, we get the comparative table following.



	Steam at 1 Lb. per Sq. Inch	Hot Water
Weight to be passed per hour	1030 lbs. steam	40,000 lbs. water (at 25° drop)
Flow pipe size	5 in. steam	$3\frac{1}{2}$ in.
Return pipe size	2 in. condense	$3\frac{1}{2}$ in.
Velocity of flow	53 ft./sec.	3 ft./sec.
Heat input to steam or water contents of flow pipe heated from cold (50°)	6400 B.T.U.'s	400,000 B.T.U.'s (water raised to 150°)

(The above figures are approximate, and ignore radiation from the main.)

The great difference in time lag is clearly shown by the last figures; in fact, if the heater at *X* has an output of no more than the 1,000,000 B.T.U.'s to be passed, it will take, in the case of hot water, half an hour before the flow pipe is brought up to temperature (allowing for radiation loss), and a further half hour before the return is fully heated. With steam only a few minutes are necessary.

The various methods of using steam for heating buildings have been mentioned in Chapter IV, to which reference should be made.

The chief systems are as follows:

Low-Pressure Steam, Single Pipe System (Fig. 186)—The steam at 2 to 5 lbs. per sq. inch pressure enters the radiators, pushing the air before it through the air valves. The steam is condensed and falls to the bottom of

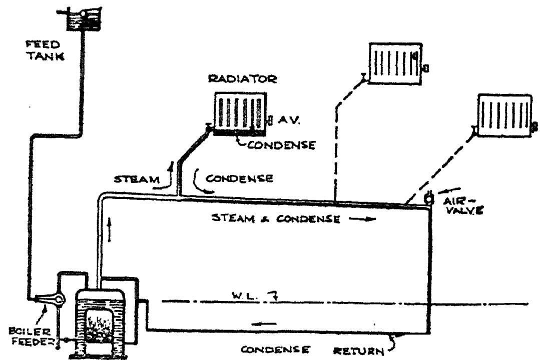


FIG. 186.—Low-Pressure Steam Heating, Single Pipe System.

the radiators, finding its way out through the same valve and pipe by which it entered. From here it returns to the boiler by flowing with the steam in the main to the far end, and back in the 'wet return'.

The system is easy and cheap to install, but is often noisy in operation and gives trouble if the radiator valves are left partly opened, due to water binding in the radiators. The piping may also be arranged as a drop system for serving a multi-storey building.

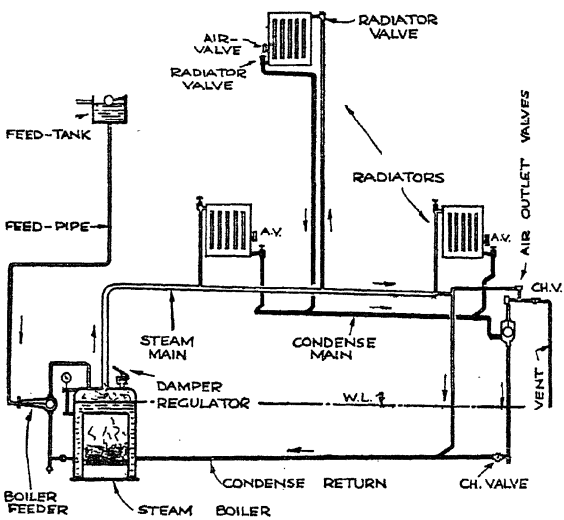


FIG. 187.—Low-Pressure Steam Heating—Two Pipe System.

Low-Pressure Steam, Two Pipe System (Fig. 187)—Steam at the same pressure as with the one pipe system is delivered to the radiators, air being discharged as before. After condensing, it passes out through another valve to a separate return main; this return may be either 'wet' as shown, or 'dry', in which case it is above water level until close to the boiler, where it drops and enters at the bottom, as with the 'wet' type.

The system may also be applied as a drop system for high buildings. Whilst more consistent in working than the single pipe system it has numerous shortcomings.

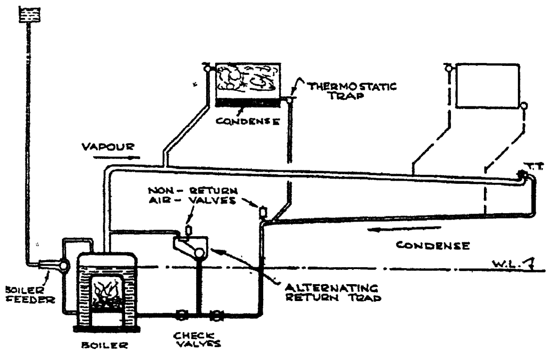


FIG. 188 (a).—Vapour Heating System.

Vapour System (Fig. 188 (*a*))—Steam at a few ounces per sq. inch pressure is delivered to the radiators through a system of pipes. The air, being more dense than the steam, falls to the bottom of the radiators and finds its exit with the water of condensation through a thermostatic trap into the return main. The latter may again be either 'wet' or 'dry'.

As the condensate and air travel back in the same return main they

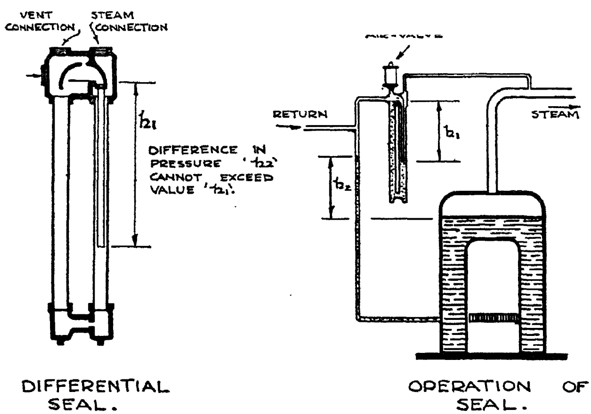


FIG. 188 (*b*).—Differential Seal for Vapour System.

must be separated out before re-entering the boiler. This may be accomplished by a variety of patented devices, one of which, termed an *alternating return trap*, is shown in the diagram. It fills up with water, the air meantime escaping by the vent; when the water level reaches a certain point the float quickly closes the vent and opens a valve in the equalizing pipe to the boiler. The water is then subjected to the same pressure as in the boiler, but, being at a higher level, flows down through the check valve next to the boiler until the trap is almost empty, when the cycle repeats.

A second device, shown in Fig. 188 (*b*), is the *differential seal* which balances the boiler pressure against the return condense so as to allow the latter to flow back at all times. The air escapes via the thermostatic vent valve, which closes when vapour reaches it from the boiler.

By using suitable fittings this system may be made to work with a banked fire at pressures below atmosphere for periods of four to eight hours, depending on the amount of inward air leaks into the system.

An advantage claimed for this system is that the radiator valves may be used as 'modulators' (see Fig. 188 (*c*)) for regulating the amount of heat in the room. When shut half off, only half the radiator may be filled with vapour.

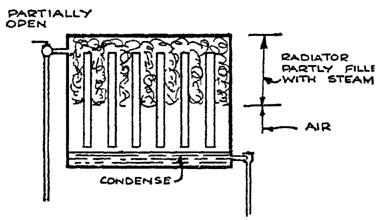


FIG. 188 (*c*).—Modulating Effect of Radiator Valve with Vapour System.

An alternative version of this system dispenses with thermostatic radiator traps, and limits the amount of steam admitted by means of a disc in the valve union (see Fig. 196 (*b*)). The disc has an orifice in the centre calibrated for various radiator sizes and admits no more steam than will condense completely before it reaches the outlet. In order to be sure of this, the heating surface is generally oversized by 10 or 20 per cent. above normal.

A further alternative version is the open type vapour system, in which the return line is left open to atmosphere and in which no partial vacuum is possible. This is obviously inferior to the above.

Sub-Atmospheric or Differential System—This system (see Fig. 189) takes advantage of the reduction of temperature of steam at sub-atmospheric pressures. The boiler operates at a maximum of 2 lbs. per sq. in. gauge pressure. The steam passes through a modulating valve controlled by hand or thermostatically according to the weather, and thence passes to the radiators or convectors. The latter are fitted with an orifice disc (see page 338) or needle valve to limit the amount of steam to that required to keep the radiator just heated all over at lowest working pressure.

Each radiator or convector is fitted with a steam trap, and the condense return lines connect to a vacuum pump which exhausts the air,

vapour and water. The latter is returned to the boiler, and the air is passed to atmosphere. The pump is capable of producing a vacuum of 25 ins. mercury, corresponding to a steam temperature of 133° F., but the vacuum is controlled by the differential controller so as to be slightly lower than

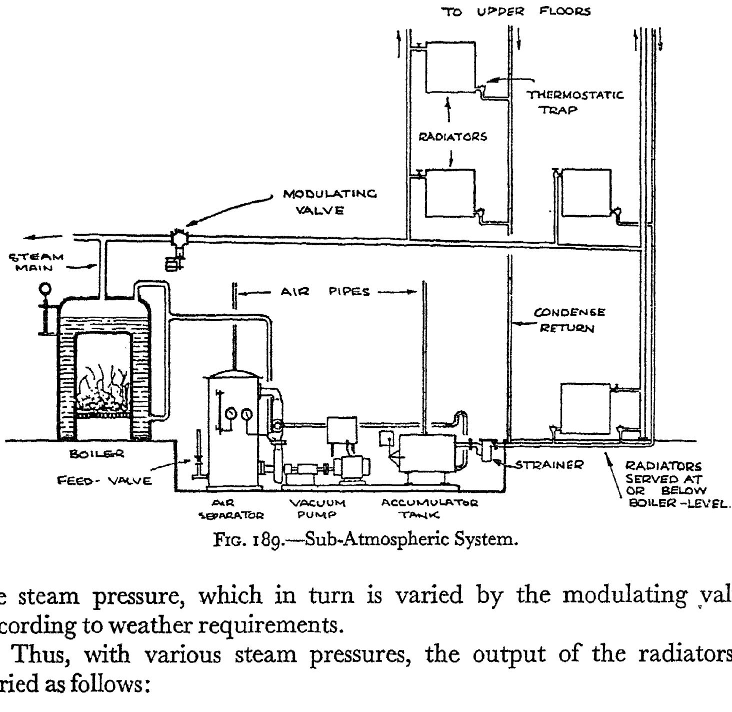


FIG. 189.—Sub-Atmospheric System.

the steam pressure, which in turn is varied by the modulating valve, according to weather requirements.

Thus, with various steam pressures, the output of the radiators is varied as follows:

Steam Pressure	Temp. °F.	Return Vacuum	B.T.U.'s per Sq. Ft. Radiation (Air at 70° F.)
2 lbs. gauge -	218°	3″–7″	240
10″ vacuum -	192°	11″–12″	198
15″ ,,	179°	16″–17″	176
23″ ,, -	150°	24″	122
25″ ,, -	133°	25″	102

Below this, as the modulating valve continues to throttle the steam in milder weather, the radiators become only partially filled with steam at 133°, so that their output is further reduced, until, with the modulating valve shut completely off, no heat is delivered. The purpose of the orifice plates in the radiator inlets is to apportion the restricted steam supply evenly throughout the system.

There are various refinements in the controls, put forward by the makers, which need not be discussed here. The steam supply may be from

a central system and not from a boiler; in such case a reducing valve to give 2 lbs. per sq. in. would be placed before the modulating valve.

It will be seen that very complete control is effected over a wide range of temperature, giving results comparable with a hot water system. Considerable fuel economies are possible over an ordinary low-pressure steam system.

Vacuum System—In the 'straight' vacuum system the same range of temperature cannot be obtained. Steam is supplied at a constant pressure of 1 to 5 lbs. per sq. in. gauge, and the vacuum exists only on the outlet side of the traps. The same type of vacuum pump may be used, but runs at a steady suction of 10 to 11 ins. mercury. It serves to exhaust the air and water effectively and quickly, and is much used with unit heater and similar systems, where control of steam temperature is not required. Otherwise it is inferior to the sub-atmospheric system.

A variation of this is a system in which each radiator or convector is fitted with an individual thermostatic control valve. This valve throttles the admission of steam to each heating device, so that a vacuum can exist therein with consequent lowering of output as before. Sometimes the steam trap is omitted altogether, the thermostat being so adjusted that, when full open, all steam is condensed before reaching the outlet.

Exhaust Steam Systems—When exhaust steam is available from engines the vacuum system may be used, condensing the steam direct without the calorifiers necessary with hot-water heating. The connections are then as shown in Fig. 190 (p. 314). The vacuum pump may be of centrifugal type with receiver, driven by motor or small steam turbine, exhausting into the low-pressure line. When this is done the power consumption of the pump costs almost nothing.

Alternatively, a reciprocating direct-acting steam pump may be used without a receiver, as shown in the figure. Such pumps handle the condense, vapour and air in the same cylinder and need to be of large size working at very low speed. On each exhaust stroke there is a risk of the condense flashing back into steam as the pressure in the cylinder is reduced. This may be overcome by fitting a cold-water connection immediately on the pump suction, supplying just enough to prevent re-evaporation.

Reciprocating vacuum pumps cannot be used to feed direct into boilers, as the air is still entrained with the water. A separating vessel or hot well is necessary to receive the discharge, followed by a boiler feed pump of ordinary type as shown in Fig. 190. This type of pump is only used where pressure steam is available (50 lbs. per sq. in. or over), and in such cases the hot well or boiler feed tank is generally necessary for other reasons.

A diaphragm-operated vacuum controller may be fitted as shown to maintain a steady vacuum.

The advantages of using exhaust steam from power plants are dealt with in Chapter XX.

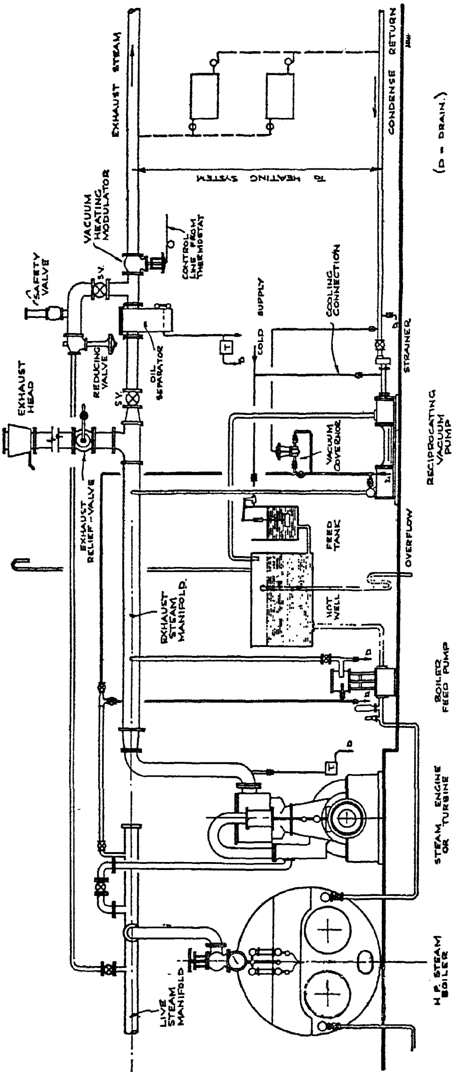


FIG. 190.—Vacuum Steam System operated from Exhaust Steam Supply.

Low, Medium and High-Pressure Steam for Heating—Steam at pressures from 5 lbs. per sq. in. upwards is commonly used in the following types of buildings:

Factories and Workshops, for heating by means of unit heaters, exposed pipe coils, radiators or convectors. Alternatively steam may be used for the heat supply to Plenum warm air systems. Steam is also often required for process plant.

Aircraft Hangars and Sheds, for heating by means of unit heaters.

Canteens, for heating by means of unit heaters or convectors, or for supplying heat to ventilation air systems and to kitchens.

Theatres and Cinemas, for supplying heat to ventilation systems, and to hot-water radiator systems through calorifiers.

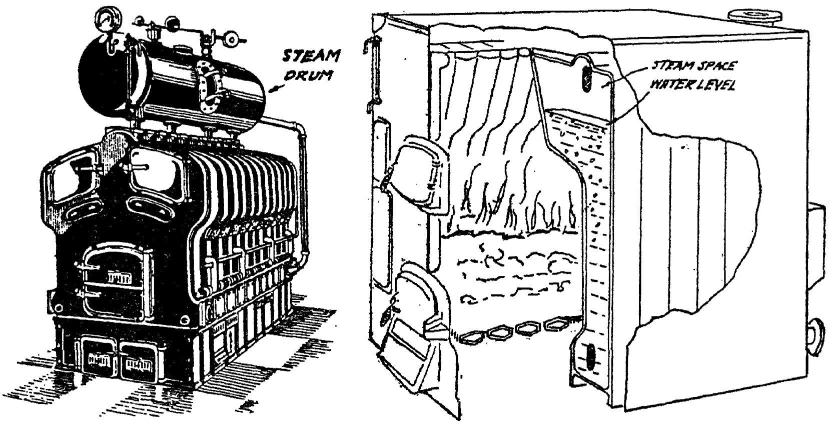
Hotels and Restaurants, for supplying steam for cooking and laundry, and through calorifiers for hot water for radiators and for hot-water supply.

Hospitals and Institutions, for supply of heat for all purposes either direct or through calorifiers.

Laundries, for supplying steam for washing, drying, etc.

Swimming Baths and Bath Houses, for heating water through calorifiers.

Low-Pressure System—Where heating and hot-water supply only is required in a single building or small group of buildings, low-pressure steam boilers of the cast iron type may be used, such as are shown in



Beeston 'Colonel'. Ideal 'A'.

FIG. 191.—Low-Pressure Cast Iron Steam Boilers.

Fig. 191. These boilers work at pressures up to 15 lbs. per sq. in., and are economical in first cost. They must have a closed system with no steam taken off which is not returned as condense, as any make-up water will cause deposits in the boiler, which it is not practicable to clean out.

There is, furthermore, a limit to the distance which low-pressure steam may be carried in pipes for economical design, owing to the pressure drop in transit, so that extensive runs are not desirable.

It will be noted that the steam space is formed in the top of the boiler or in the steam drum and is limited in capacity, so that sudden heavy demands may be accompanied by a rapid drop in pressure, and unsatisfactory distribution. This sometimes occurs where these boilers serve unit heaters on thermostatic control. Some means are desirable for limiting the number of units which can be switched on simultaneously.

Medium Pressure Steam—For more extensive heating systems, and for all cases where steam is used direct, not being returned as condense (as in some kitchen apparatus, laundries, process plant, etc.), a pressure at the boiler of 25 to 80 lbs. per sq. in. is usual.

The boilers for such purposes are of steel of 'shell' type, such as the Cornish, Lancashire, Economic, Super-Economic, or vertical, as described on pp. 96-101.

After leaving the boiler the steam may be reduced in pressure by means of pressure-reducing valves for various purposes as required, but so far as the heating system is concerned it may be used at full boiler pressure, to serve pipe coils, unit heaters, air heaters and convectors, provided they are designed to suit. Cast iron radiators are usually limited to about 15 lbs. per sq. in.

High-Pressure Steam—For more extensive systems of steam distribution, as over large factory or institution sites, a higher pressure still is generally adopted, such as 100, 125, 150 or 200 lbs. per sq. in. These pressures are often necessary for process plant in factories.

Shell type boilers are suitable for pressures up to 250 lbs. per sq. in., and for steam loads of 15,000 to 20,000 lbs. steam per hour per boiler. Thus, for higher pressures and greater loads than, say, 100,000 to 150,000 lbs. per hour, for which up to ten boilers might be required, it is desirable to adopt the water-tube type boiler.

Plate XIX shows a range of four water-tube boilers working at 125 lbs. per sq. in., output 40,000 lbs. per hour each, connected to a large heating system installed under the direction of the Authors.

Water-tube boilers are constructed for any pressure up to 2,000 lbs. per sq. in.; the higher pressures over 250 lbs. per sq. in. are required where power is generated by steam engines or turbines, in order to achieve higher efficiency. Such pressures are generally accompanied by a considerable degree of superheat, but are outside the realm of normal heating practice, and need not be considered here.

Superheating of steam is not usually resorted to for straight heating (as apart from generating stations), though it has advantages in the transmission of heat over long distances, as for example in District Heating. The question of superheating is dealt with more fully in the Authors' companion volume on *District Heating*.

Choice of Steam Pressure—The pressure selected for the heating system will be affected by the following considerations:

(*a*) For cast iron radiators, 15 lbs. per sq. in. is the maximum.

(*b*) Unit heaters, air heaters, convectors and pipe coils can be used at any normal pressure. In the case of unit heaters and convectors, a lower pressure permits the use of lower final air temperatures, generally a desirable feature. A pressure of 15 to 30 lbs. per sq. in. is often selected.

(*c*) The lower the pressure the larger the pipes necessary for a given

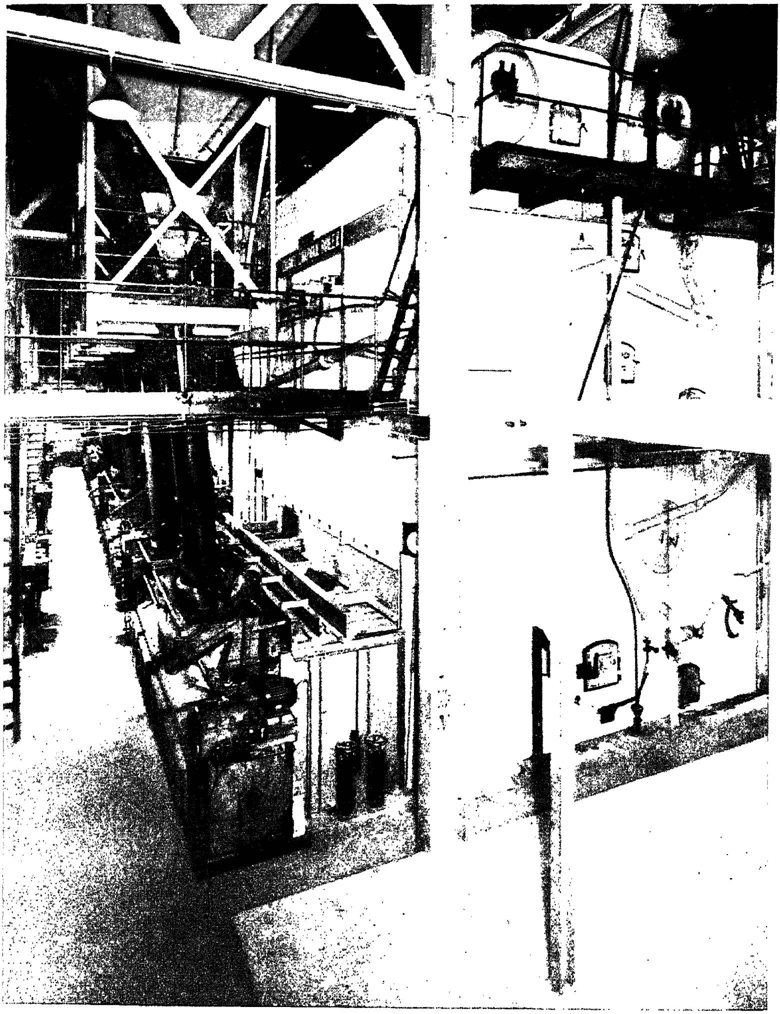


Plate XIX. Range of four Water-tube boilers for heating of large Government Depot *(see p. 316)*

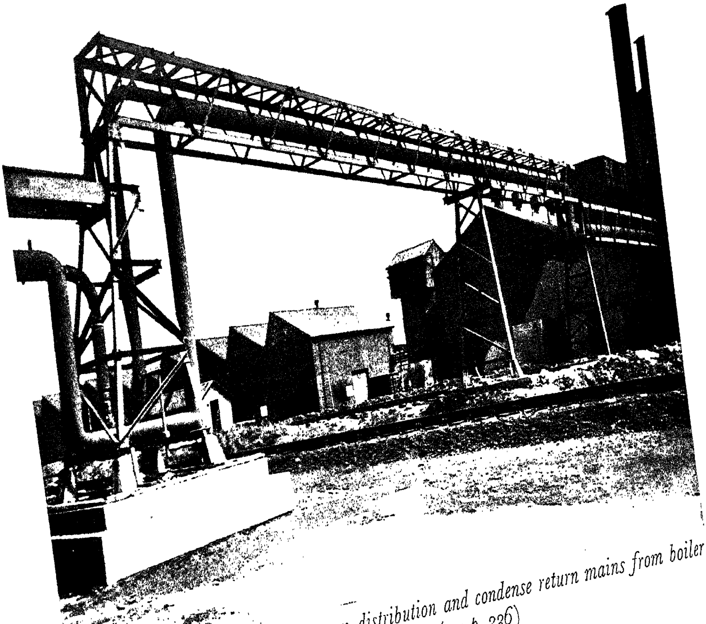


Plate XX. Showing high pressure steam distribution and condense return mains from boiler house to factory buildings *(see p. 336)*

load, hence for larger systems a higher pressure is desirable for economy.

(*d*) The higher the pressure and temperature the greater the losses due to radiation.

(*e*) Where a vacuum system is adopted, steam at 5 lbs. per sq. in. will be considered a maximum.

(*f*) Re-evaporation losses from traps are reduced at lower pressures. This is referred to later under Condense Return.

The pressure of the heating system need not be the same as the boiler pressure. There are reasons why it is preferable to have a high boiler pressure serving the system at a lower pressure through a reducing valve. Among these reasons are the following :

(*a*) The steam will be drier after reduction, due to any wetness being re-evaporated into steam at the lower pressure. All apparatus is designed for dry saturated steam, and its output suffers if wet steam is supplied.

(*b*) The effect of sudden variation of load does not react on the boiler pressure so greatly if the latter is kept at, say, 80 lbs. per sq. in. and the condensing apparatus takes steam through a reducing valve at, say, 30 lbs. per sq. in. This is because the heat storage in the boiler is materially increased at the higher pressure, not only due to the accumulated steam of greater density, but on account of the stored hot water at steam temperature. Any slight drop in pressure in the boiler allows this to evaporate and supply steam. With boilers of small steam space, such as Economic, vertical and water-tube types, this is important.

(*c*) A further disadvantage of using steam straight from a boiler at low pressure is that the rate of steam release from the water surface at low pressure, in terms of volume, is much greater than at high pressure. At low pressures this release may be so rapid as to carry water over into the steam mains, a feature known as 'priming'. When this occurs serious troubles with steam main drainage commence, leading to water hammer, and sometimes a disappearance of the water line in the boiler.

(*d*) Where an economizer is installed, as with a Lancashire boiler, a higher pressure enables a higher efficiency to be obtained, because the economizer outlet can be raised to a higher temperature before ebullition occurs, hence more heat can be picked up from the flue gases. The normal economizer outlet is kept some 30° below the boiling point corresponding to the pressure, in order to avoid steam formation. There are some types where steaming may occur without harm, but even so the efficiency is always limited by the boiler pressure.

CONDENSE RETURN

When the steam has condensed into water in the heating apparatus a means for its escape has to be provided. As the water is drained off the steam follows, and will blow to waste if unchecked. To prevent this, a device known as a *steam trap* is always installed on the outlet of each piece of apparatus (with certain special exceptions). Types of traps are dealt with later.

On leaving the trap the water will be at the same temperature as the steam. It will be obvious that the water cannot exist as such above 212° F. in free air, and therefore, if the steam pressure is above atmospheric, a portion of the condensate will re-evaporate into steam, drawing the latent heat necessary for this operation from the remaining bulk of water which is consequently lowered in temperature to 212° F. or below. The amount of re-evaporation can be calculated from steam tables and depends on the pressure. The higher the pressure the greater the loss due to re-evaporation.

If the condensate is run to a drain, fresh cold water will have to be supplied to the boiler to replace it, and all the sensible heat from the temperature of the steam down to the temperature of the boiler feed will be wasted. If it is returned, both heat and water are saved. Thus, it is generally found desirable to return all condensate possible to the boiler house, where it may be fed back into the boiler.

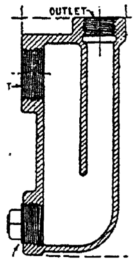
In a long condense line the re-evaporated vapour may be re-condensed, due to the radiation from the line. If the piping is all inside the building, this does not represent a loss, as the heat given up is useful in warming the building, but if the condense main is not in the building to be warmed the radiated heat is lost.

One method of reclaiming this re-evaporation loss, and removing some of the sensible heat, is to pass the condensate through some device which can remove the heat and apply it usefully. Thus, in a unit heater system, one unit can take the whole condensate from one building and cool it down before its return, so usefully heating the building. The same can be done with a pre-heating coil in a hot-water supply calorifier. A further point is that water at about 180° is preferable to water at 212° for boiler feed if pumps are used, due to the flashing into steam which may occur at the latter temperature at the reduced pressure of the pump suction. This can be very troublesome, and can only be cured by dropping the temperature by injection of cold water to the suction or by raising the condense tank well above the pump.

Another system designed to overcome the re-evaporation loss problem is one in which the steam is used in a series of stages, generally three. The heating system, such as a range of unit heaters, is divided into sections. The first receives steam at full boiler pressure. The condensate is allowed to re-evaporate at a lower pressure, being expelled by a steam ejector, and is used in the second section. The same occurs in the third stage, so that the fullest use is made of the heat in the steam.

Methods of returning the condensate from the traps to the boiler house are:

(*a*) Gravity return—possible where levels permit.
(*b*) By elevating condensate by lifting trap or super-lifting trap, then by gravity.
(*c*) By condense-return pumping unit.
(*d*) By vacuum pump. (When this is used the condensate may be lifted, provided the pump is kept running continuously. Each lift should not exceed 5 ft., but there may be a series of steps in cascade form. At the base of each lift a lift-fitting is necessary.)



Lift Fitting (Dunham).

These devices are described later. All condense lines should have a fall in the direction of flow of about 1 in. in 20 ft.

Methods of feeding the return water and make-up water to the boiler are:

(*e*) By gravity—on small systems only. In this case the make-up water is admitted either by hand or by 'banjo' feeder (see later).
(*f*) By alternating trap (described later).
(*g*) By feed pump from a return tank or hot well. The pump may be either steam-driven or electrically-driven. Steam-driven pumps are either of reciprocating type or turbine-driven centrifugal type. The latter are used for large plants, over, say, 50,000 lbs. steam per hour.
Electrically-driven pumps are generally centrifugal.
(*h*) When methods (*c*) and (*d*) above are used for condensate return they may deliver direct into the boiler if this is at low pressure. If at a higher pressure they generally deliver to the hot-well and are pumped to the boiler as in (*g*).

AIR IN STEAM SYSTEMS

When a system is first started up it is full of air. As steam is generated and passes along the pipes this air goes before it and is compressed. Were it to remain, no further steam would be admitted and the system would not function. When the system is at work normally, air continues to collect as it is dissolved in the water and comes out in the steam.

The removal of air from the system can be effected by opening a cock at the bottom of each radiator, etc. (air being denser than steam falls to the bottom), or by an automatic air valve, closing by thermal expansion when the steam arrives. With thermostatic steam traps of most present-day types the air passes the trap into the return line, and no separate

air valve is necessary. Certain traps of the bucket type do not allow air to pass.

The ends of steam mains require provision for air removal either by suitable trap or by automatic vent; similarly for the condense return mains on a closed system near the boiler.

What has been said above about air removal applies particularly to low-pressure steam. With the vacuum system the air is withdrawn by the pump. With medium and high-pressure steam, air does not present a problem, as it is greatly compressed and so passes out with the water, when a trap opens, in the form of small bubbles. No special provision is necessary for its removal apart from the initial clearance of the system by blowing it from an open end.

CORROSION IN CONDENSE LINES

One of the bugbears of a steam system is the corrosion which occurs in the condense return lines. This is due chiefly to dissolved oxygen and carbon dioxide. On extensive systems some treatment is often applied to the feed water, such as by the addition of tannin, which is an oxygen inhibitor.

If, however, the condense lines are made of copper instead of steel, the trouble is overcome, though at some extra cost. Larger sizes, say, 3 ins. and over, may be made of cast iron flanged, which again is more resistant to corrosion than steel. Wrought iron has also been much used, but is becoming increasingly short in supply. Some engineers advocate galvanized return lines, but this is not always a success, as sometimes there is a small amount of chlorine in the steam, which dissolves in the condense and attacks the zinc.

With copper pipes the fittings would be of gun metal, and with ferrous piping of cast iron. Malleable iron fittings on condense seem to be very prone to rapid attack.

In the case of the Vacuum System, one of the advantages claimed is a reduction in the corrosion of condense lines, due to the rapid evacuation of the air.

EMISSION FROM STEAM-HEATED SURFACES

The emission from any form of steam heating apparatus depends on the temperature difference between the steam and air of the room being heated. Temperatures of steam may be taken from Table LX on p. 351, e.g. at 1 lb. per sq. in. the steam temperature is 215° F. With air at 60° the difference will be 155° F.

Pipes and Radiators—Table LVII (p. 327-8) gives transmission coefficients for various types of heating surface, also the total emission for the case of steam at 1 lb. pressure, air at 60°.

UNIT HEATERS*

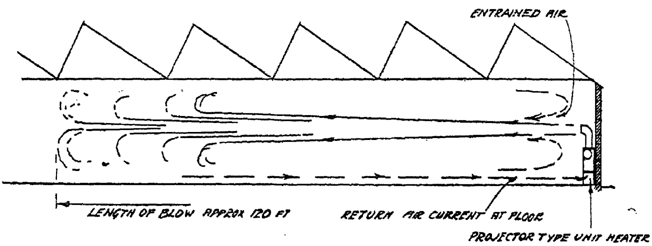
As explained earlier, Unit Heaters are a satisfactory and economical method of heating large spaces. Various types were shown on page 89, i.e. horizontal discharge, vertical discharge, and projector type.

Ratings of unit heaters depend on steam pressure, air temperature, and air speed. For complete information it is necessary to refer to makers' catalogues.

The sizes of the *horizontal type* (Fig. 41 (*a*)) range from about 16,000 to 300,000 B.T.U.'s per hour. The mounting height requires to be checked with the makers. For low heights such as 8-12 ft. a low air speed is desirable with a greater number of small heaters. For mounting heights 15 to 25 ft. a higher air speed is desirable to deliver the warm air to the floor. The air is deflected by the louvres downwards at an angle of from 15 to 45°.

The *vertical type* (Fig. 41 (*b*)) is designed for mounting heights of 20 to 35 ft. above floor, and the sizes range from about 100,000 to 600,000 B.T.U.'s per hour. These units are used where the mounting height is too great for the horizontal type. High duty fans are frequently fitted to overcome the natural tendency of the heated air to rise. Various makes adopt different methods to protect the motor from being overheated by convection from the heating battery when the fan is stopped, and again different delivery deflectors to prevent the heated air spreading before reaching floor level.

The *projector type* (Fig. 41 (*c*)) is most suitable for floor mounting, though it may be fixed overhead. Sizes range from about 100,000 to 1,000,000 B.T.U.'s per hour. When fixed near the floor, air is drawn in at floor level over a heating battery, and is delivered by a series of cased fans running on a common shaft, through nozzles 10 to 12 ft. from the floor, at a velocity of about 2000 ft. per minute.



Air Circulation with 'Projection' Type Unit Heaters.

The length of blow† is about 120 ft. Due to the high speed of dis-

* Unit heaters are dealt with in detail here under 'Steam Heating', though they may equally be used with hot water, which, for a large installation, would preferably be at high temperature (see Chapter XV on high-pressure hot water). The horizontal type of unit is also available for gas and electricity (see Chapters XII and XIII).

† This is generally taken as the length to a point where the air speed has dropped to about 40 ft. per min.

charge a considerable volume of entrained air is induced into the stream. The action of returning the air at floor level promotes an efficient circulation at the working plane. When the final air temperature is kept at about 110° there is little tendency for the air to rise at the end of its travel, so that roof temperatures tend to be slightly lower than floor temperatures, which, of course, reduces heat losses.

It will be seen that unit heaters of this type may be used to cover a wide area effectively, no matter what the height of the building. They are more easily accessible for maintenance than overhead types, but their use is restricted to cases where there are no internal partitions or obstructions (such as aircraft in an aircraft shed) to interfere with the free discharge of air.

Final Temperature—With any type of unit heater it is preferable to limit the final air temperature to about 110°-120° F. If higher temperatures are used, such as 130° or 140° the buoyancy of the air becomes so great that it tends to rise to the roof as soon as the velocity is dropped, without mixing with the room air.

The temperature rise can easily be calculated, thus:

e.g.	Unit heater rating - - - -	120,000 B.T.U./hr.
	Fan volume - - - - -	2,000 cu.ft./min.
	Entering air from room - - -	60°

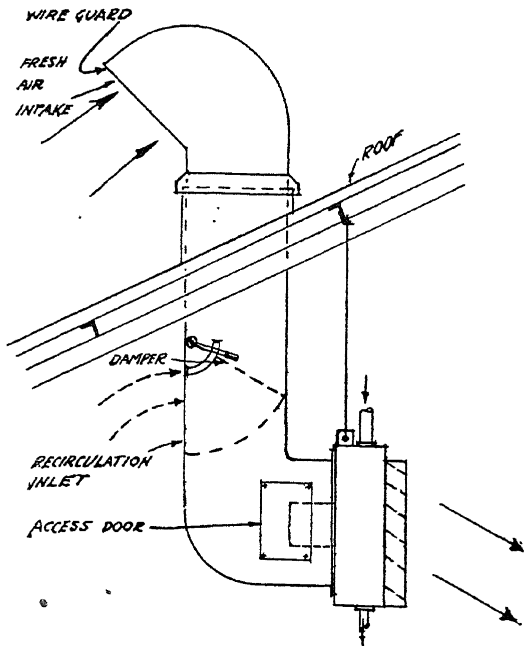
$$\text{Temperature rise} = \frac{120,000}{2000 \times 60 \times \cdot 019} = 53^\circ.$$

The final temperature will then be $60 + 53 = 113°$ F.

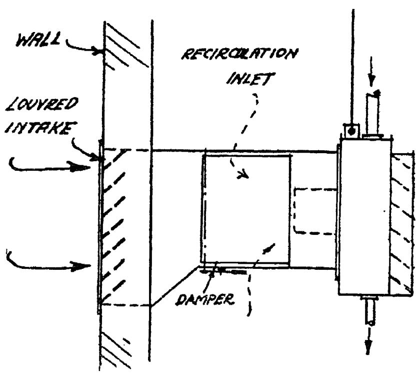
It is particularly important to check the final outlet temperature where low speed fans are used, as the air volume is then much reduced and the temperature may be unduly high.

Noise—Unit heaters are not suitable for places where quiet is essential. Low speed units (730 r.p.m.) are permissible in normal factory offices, canteens, etc., where the sound level is already appreciable. Normal (960 r.p.m.) and high speed (1440 r.p.m.) are usual in noisier situations. The projector type must be considered only where noise is unimportant.

Fresh Air Inlets—It is often necessary to provide artificial inlet ventilation to a building where unit heaters are used. This may conveniently be done by connecting the suction side of the unit by means of a duct to external air, terminating either with a roof cowl or with a louvred opening in the wall as in Fig. 192. The basis of design must take into account the volume of fresh air introduced, and that the unit inlet will be at 30° or so in cold weather. This, of course, affects the rating of the unit. When some units in a building are provided with fresh air inlets and some are recirculating, it is desirable that their final air temperatures should be as nearly the same as possible.



Unit Heater with Fresh Air Intake through Roof.



Unit Heater with Fresh Air Intake through Wall.

FIG. 192.

It is sometimes arranged that the fresh air inlet is controlled by a damper, so that in cold weather the unit may be made to recirculate.

It should be noted that the battery of a unit connected by vertical ducting to a roof cowl will induce a strong reverse air current when the fan

is stopped, with risk of overheating of the motor. It is generally arranged that these units should not be under thermostatic control by switching off the fan. If they are switched off, the steam should be shut off also.

Layout of Unit Heaters—There are many theories as to the correct method of arranging overhead units within a given building, but in fact there is no one correct method. The Authors have used many different types of layout with equal success, and it would appear that, provided the air streams are directed towards the wall surfaces and areas of greatest heat loss, and do not blow directly against one another, the layout may be varied within wide limits to suit piping runs and other considerations.

The layouts shown in Fig. 193 (*a*), all of which are actual examples, will illustrate this point.

The layout of projector type units is dependent on where floor space can be spared for them. They will generally be located around the external walls, or down the centre of a large shop on the line of stancheons. The nozzles may be turned to blow in any direction required. Large units have three or four nozzles, each of which may be set in a different direction.

Thermostatic Control—The simplest and most satisfactory method of control of unit heaters is the switching on and off of the fan by electrical thermostat. Without the fan running, little heat is emitted.

Where one or two units only occur in one room there would be a thermostat to each: where a number occur, as in a large shed, they may be controlled in groups of four or six. A contactor will then be necessary for 3-phase current, the thermostat making and breaking the coil circuit only.

A hand switch is desirable to short circuit the thermostat so that the units may be run in summer without heat on, to provide air movement.

Motors—The size of motors for unit heaters varies from about $\frac{1}{16}$ to 1 h.p. for horizontal and vertical types, and up to 5 h.p. for the projector type. Where alternating current is available single phase motors are usually of the 'Capacitor start-run'* type which have no brush gear to give trouble, but sometimes the condensers break down, due to overheating from the radiant heat. Where it is possible to use three-phase current it is preferable, though some protection against single-phasing seems to be desirable, but for small fans the cost of protection is nearly equal to that of the motor. Faulty contactors or wiring may cause burn-outs through this taking place. Motors are usually best totally enclosed, and the bearings should be suitable for long running without attention.

Margin—The required effect of unit heaters is sometimes not achieved in practice, due to steam pressure drop, dirt on the fins of the heater, direct escape of warm air through doors before mixing, unsatisfactory distribution and other causes. It is customary to make the installed load somewhat

* In this, part of the field winding is connected through a condenser to give an 'out of phase' effect for starting. The condenser remains in circuit whilst running. The capacitor-start induction run motor has a centrifugal switch for cutting out the condenser after starting, but the switch requires periodical attention.

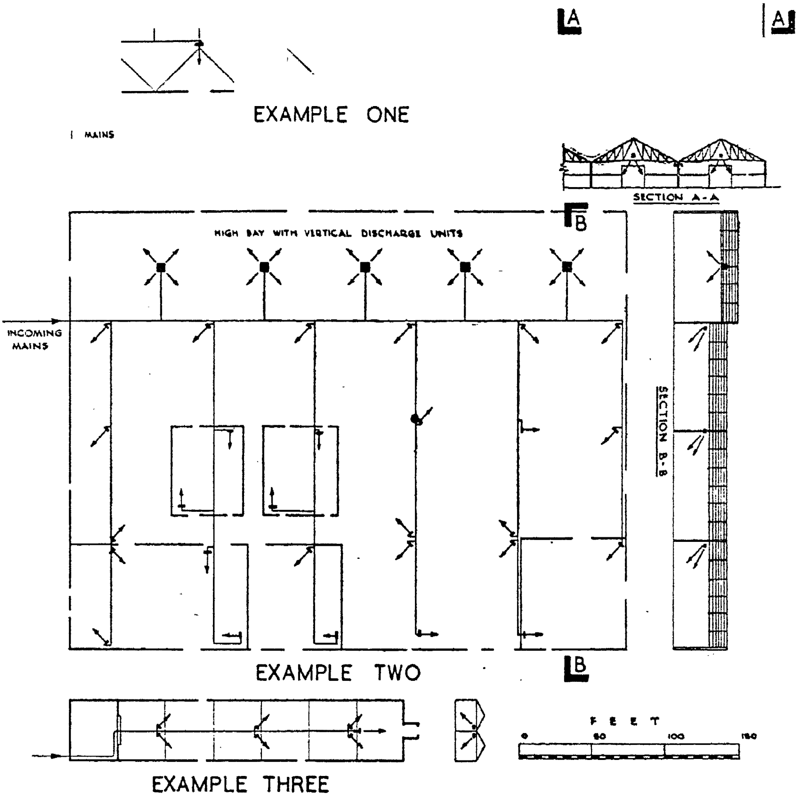


FIG. 193 (*a*).—Typical Unit Heater Layouts.
Example One—Storeshed.
„ Two—Workshops.
„ Three—Canteen.

he calculated heat losses. A margin will furthermore allow the control to take charge even during coldest weather when the otherwise be running continuously.

ch a margin must be purely empirical and is generally taken 15 to 30 per cent. of the heat losses. This margin will also be available for quick heating up, and the piping and boiler power should be installed to clude for it.

ining Connections—The large concentrated steam loads of unit heaters for care in the piping connections. The steam inlet requires a valve, eferably of the fullway gate type. The condense outlet requires a trap capable of passing large volumes of condense rapidly, as on heating up a cold building the condensation rate will be much higher than when warm.

For small units, up to about 100,000 B.T.U./hr. on low pressure or vacuum steam, a thermostatic trap will suffice, over this a float and tic type is necessary.

edium and high pressure units an inverted bucket and certain of float trap are suitable.

nions are necessary for disconnection of steam and condense.

Fig. 193 (*b*) shows a typical arrangement of connections.

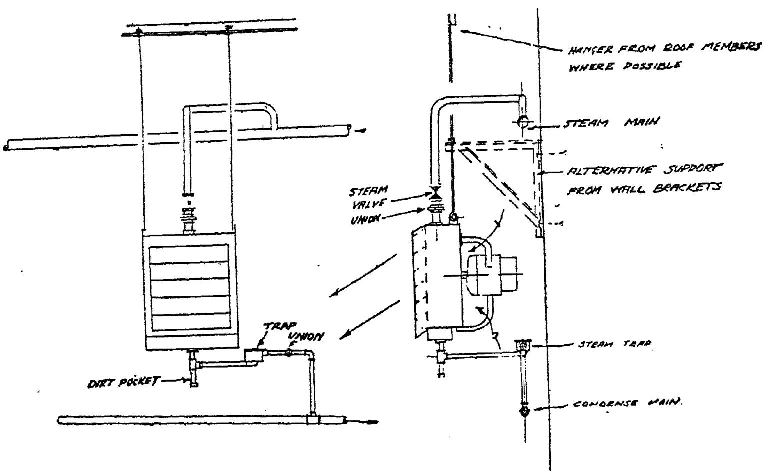


Fig. 193 (*b*).—Typical Unit Heater Connections.

CONVECTORS

Convectors of the same general type as those already described (see pp. 83 and 179) may be used with steam as the heating medium, and while it is physically possible to operate them with the same range of pressures as, for instance, unit heaters, it is not advisable to use pressures above about 15 lbs. for the following reasons.

(*a*) The high temperatures associated with steam heating give rise to a feeling of stuffiness and to the 'burning' of dust particles in the air with a characteristic smell.

(*b*) Convectors are used generally in relatively small spaces as compared with other forms of steam heating, and with high pressures the control is not fine enough to allow the occupants to maintain comfortable conditions.

Steam convectors are an economical way of heating factory offices, particularly where these occur singly or in small groups over a wide area, of factories where steam is already being distributed for heating of the main workshops; but for larger compact office layouts the Authors generally prefer to install a small steam-water calorifier serving hot-water radiators, either by gravity circulation or with a small pump.

Convectors may also be used with low-pressure or sub-atmospheric steam systems, for ordinary domestic heating, and are so used on a large scale in America, where steam domestic heating is the rule rather than the exception, as discussed at the beginning of the chapter. For such applications the fittings, etc., are the same as for a radiator (see p. 338).

Rating of Convectors—Table XXXIVB on page 180 gives the equivalent sq. feet of heating surface for various sizes and heights of convectors, and to find the emission of a given unit it is only necessary to multiply the surface by the appropriate emission per sq. foot from Table LVIIC, as already explained.

TABLE LVIIA

EMISSION FROM STEAM-HEATED SURFACES

Radiators Type 'Ideal'	Coefficient B.T.U./Sq. Ft./Hr./ 1° Diff. Steam-Air	B.T.U./Sq. Ft./Hr. Steam 215° Air 60° } 155° Diff.
Neo-Classic No. 2	2·11	327
,, ,, 4	1·94	300
,, ,, 6	1·82	282
,, Window	1·79	278
Hospital 3″	2·11	327
,, 5¾″	1·79	278
,, 7¼″	1·70	264
Classic Wall	1·94	300
Plain Wall	1·82	282
,, 1 Col.	1·82	282
,, 2 ,,	1·78	275

(Above information supplied by Ideal Radiator Co.)
The above transmissions are for radiators standing $1\frac{1}{2}$ in. or more from wall.

If enclosed in open recess deduct - - - - - - 10%
,, ,, recess with metal grille deduct - - - 20%-25%

The coefficients are for radiators of ten sections or more; for less than ten sections the values may be increased by 5%.

TABLE LVIIb

Steel Piping, Nominal Bore	External Surface: Sq. Feet per Ft. Run	Coefficient in B.T.U./Sq. Ft./Hour/1° F. Difference Steam-Air						B.T.U./Ft. Run Steam 215°, Air 60° — 155° Diff.
		at 275° Difference	at 250° Difference	at 225° Difference	at 200° Difference	at 175° Difference	at 155° Difference	
½"	0·21	3·62	3·47	3·34	3·15	3·02	2·89	94
¾"	0·28	3·52	3·37	3·22	3·12	2·92	2·81	124
1"	0·33	3·42	3·28	3·12	2·97	2·83	2·72	139
1¼"	0·43	3·33	3·19	3·04	2·89	2·76	2·63	175
1½"	0·50	3·29	3·15	3·00	2·86	2·71	2·60	205
2"	0·62	3·21	3·07	2·93	2·78	2·65	2·54	244
2½"	0·75	3·13	2·99	2·85	2·71	2·58	2·47	286
3"	0·92	3·09	2·94	2·80	2·66	2·53	2·42	345
4"	1·19	3·02	2·87	2·73	2·63	2·47	2·35	434
5"	1·46	2·95	2·82	2·68	2·54	2·41	2·30	520
6"	1·75	2·91	2·77	2·63	2·50	2·37	2·27	615
7"	2·00	2·87	2·73	2·60	2·46	2·34	2·24	695
8"	2·26	2·83	2·70	2·57	2·43	2·31	2·20	770
9"	2·52	2·80	2·67	2·54	2·41	2·28	2·18	855
10"	2·82	2·78	2·65	2·52	2·39	2·26	2·16	945
12"	3·35	2·74	2·61	2·48	3·35	2·23	2·12	1100
Example: Air at 60°, Steam at (Lbs./Sq. In. Gauge):		95 Lbs.	63 Lbs.	38 Lbs.	20 Lbs.	8 Lbs.	1 Lb.	

(a) For banks of pipes close together, reduce the total emission as follows:
For 2 rows of pipes take 80% of total emission.
For 4 " " " 66% " " " "

(b) For emission from insulated piping (lagging efficiency $E\%$) multiply above figures by $\frac{E}{100}$

TABLE LVIIc

EMISSION FROM STEAM-HEATED CONVECTORS, PER SQ. FT. EQUIVALENT HEATING SURFACE, AIR AT 60° F. Use with TABLE XXXIVb, p. 180

Steam Pressure Lbs./Sq. In. Gauge	Steam Temp. °F.	Emission per Sq. Ft. B.T.U.'s/hour
0	212	300
5	227	310
10	240	325
15	250	340
20	259	350
30	274	370
Higher Pressures not normally recommended		
40	287	385
50	298	400
60	307	410
70	316	420
80	324	430
90	331	440
100	338	450

(Information supplied by Messrs. British Trane Co. Ltd.)

PIPE SIZING FOR STEAM SYSTEMS

For the purpose of sizing steam mains and piping it is necessary to determine:

(*a*) The quantity of steam in lbs. to be conveyed per hour. This may be derived in the case of a heating system directly from the B.T.U.'s transmitted, by dividing by the latent heat at the appropriate pressure. The value at atmospheric pressure is 970 B.T.U.'s per lb., which may be taken at 1000 for approximate purposes.

In the case of kitchen or laundry apparatus the steam condensed in lbs. per hour may be arrived at by calculation if the duty is known, or from the manufacturers' data for each item.

Where the mains are lagged and comparatively short their heat loss may be ignored in estimating the steam flow. If such is not the case due allowance should be made.

(*b*) The initial pressure of the steam.

(*c*) The pressure drop permissible between the two ends of the system. This may be $\frac{1}{4}$ or $\frac{1}{2}$ lb. per sq. inch for vacuum and vapour systems, 1 to 2 lbs. for low pressure, and 5 to 20 lbs. for high pressure, depending on size of job.

(*d*) The resistances in the pipe line due to bends, tees, valves, etc.

Steam Pipes—Table LVIII (see pp. 330 and 331), may be used for estimating pipe sizes for various flows at any pressure below atmospheric or above atmospheric up to 150 lbs. absolute.

The first step is to obtain the factor Z from part C of the table for the known initial pressure and pressure drop. The pressure drop is determined by the initial pressure and size of job, as mentioned above.

In the case of a low pressure closed system the drop is limited by the permissible rise in water level in the condense return.

This factor is then divided by the single run (steam flow only) in hundreds of feet (including an arbitrary allowance of 20-30 per cent. for resistances) to give $\frac{Z}{l}$. From this the pipe size for the quantity of steam to be conveyed may be ascertained from part A of the table.

Having arrived at the first approximation of the pipe sizes, the resistances may be estimated more accurately from Part D of the table.

Taking the initial pressure as before, the pressure drop in lbs. per sq. inch for various types of fittings may be read direct. These are for $Y=1$ (Y being a function of the velocity), and must be multiplied by the actual value of Y obtained from part A by interpolating between the heavy lines.

The pressure drop for the sum of the resistances is then deducted from the permissible drop in the system, and a new value for $\frac{Z}{l}$ obtained, from which a closer sizing may be achieved.

TABLE LVIII

A. QUANTITY OF STEAM IN LBS. PER HOUR FOR VARIOUS PIPE SIZES

Z/ℓ (ℓ in 100 ft. units)	1/2"	3/4"	1"	1 1/4"	1 1/2"	2"	2 1/2"	3"	4"	5"	6"	7"	8"	9"	10"	12"	
0·1	—	—	—	—	—	65	130	205	450	820	1340	2000	2900	4000	5300	8700	Y = 10
0·2	—	—	—	30	50	100	200	300	700	1200	2000	3000	4300	5800	7800	12500	
0·3	—	—	20	40	60	120	240	400	800	1500	2500	3700	5400	7400	9700	10800	Y = 40
0·4	Y = 1 —	10	25	45	70	150	270	450	1000	1800	2900	4400	6200	8500	11300	18300	
0·6	4·5	13	30	50	80	185	340	550	1230	2000	3600	5500	7700	10500	14100	22600	
0·8	5	16	35	65	100	220	400	650	1400	2500	4100	6800	9000	12300	16200	26000	Y = 100
1·0	6	17	40	70	120	240	440	730	1600	3000	4800	7000	10000	14000	18000	31000	
2·0	8	25	60	100	170	350	650	1000	2300	4200	6800	10000	15000	20000	26000	44000	
3·0	10	30	70	120	200	450	800	1300	2900	5300	8500	13000	19000	26000	34000	56000	Y = 400
4·0	Y = 4 12	38	80	145	240	520	950	1550	3400	6100	10000	15000	21000	30000	40000	65000	
6·0	15	48	100	180	300	650	1200	2000	4200	7700	12000	19000	27000	37000	49000	80000	
8·0	18	55	120	210	340	740	1300	2200	4900	8900	14300	21700	32000	43000	57000	93000	Y = 1000
10·0	20	62	135	240	390	850	1600	2500	5600	10200	16400	25000	35000				
20·0	30	90	200	340	560	1230	2200	3600	8100	14400	26000	32000					
30·0	36	110	240	420	700	1500	2800	4500	9900	18000							
40·0	42	125	280	510	830	1800	3300	5400									
50·0	50	140	300	570	920	2000	3700	6000									
60·0	54	160	340	630	1000	2250	4100										
80·0	60	180	400	700	1250	2500											
100·0	72	210	440	800	1340												
200·0	100	300	650														
300·0	125	380															
500·0	165																

DOTTED LINE INDICATES LIMIT OF NOISELESS FLOW FOR RISERS

B. CONDENSE PIPE CAPACITIES, LBS. PER HOUR.

MEDIUM & HIGH PRESSURE	1/2"	3/4"	1"	1 1/4"	1 1/2"	2"	2 1/2"	3"	4"	5"	6"	8"
RETURN MAIN UP TO 1000 FT	120	360	800	1450	2300	5000	9000	15000	32,000	56,000	90,000	200,000
" " 1000 TO 3000 FT	–	–	200	400	1000	3000	5000	10000	20000	35000	55000	122000
LOW PRESSURE & VACUUM												
HORIZONTAL MAINS (DRY) UP TO 400'	30	85	200	400	1000	3000	5000	10,000	20,000	35,000	55,000	122,000
" " 400 TO 1000'	–	–	125	250	600	2000	3000	6000	12000	22000	35000	73000
" " 1000 TO 2000'	–	–	90	200	450	1250	2200	4000	8000	15000	25000	55000
VERTICAL RETURN DROPS	80	250	500	1250	–	–	–	–	–	–	–	–
'WET' RETURNS UP TO 400'	70	180	320	700	1000	1800	–	–	–	–	–	–
" " OVER 400'	–	70	160	340	460	860	–	–	–	–	–	–

INITIAL ABSOLUTE PRESSURE LBS./SQ.IN.	C. FACTOR Z — PRESSURE DROP IN LBS. PER SQ. IN. ¼	½	1	1½	2	3	4	5	6	7	8	9	10	12	15	20	D. * PRESSURE DROP FOR FITTINGS LBS. PER SQ.IN. SHORT BEND	LONG BEND	GLOBE OR ANGLE VALVE	TEE BRANCH
6	0·25	0·50	0·96														·0134	·0067	·060	·010
8	0·34	0·66	1·28	1·86	2·40	3·40											·0102	·0051	·046	·0076
10	0·42	0·82	1·60	2·34	3·04	4·32	5·44										·0084	·0042	·038	·0063
12	0·47	0·90	1·90	2·80	3·60	5·20	6·70										·0070	·0035	·031	·0053
14	0·63	1·20	2·30	3·30	4·40	6·30	8·00										·0060	·0030	·027	·0045
16	0·80	1·35	2·50	3·60	4·80	7·10	9·20										·0054	·0027	·024	·0040
18	1·00	1·70	3·0	4·20	5·50	8·00	10·3										·0048	·0024	·022	·0036
20	—	1·85	3·1	4·40	6·00	9·00	11·5	14·0	16·3								·0044	·0022	·020	·003[illegible]
25	—	—	4·0	5·60	7·50	11·0	14·5	18·0	21·0	24·0	27·0	29·5	32·0				·0036	·0018	·016	·002[illegible]
30	—	—	4·5	6·50	9·00	13·5	17·5	21·5	29·5	29·0	32·5	36·0	39·5	46·5	63·5		·0030	·0015	·014	·002[illegible]
40	—	—	6·0	8·20	12·0	18·0	23·6	29·0	34·5	39·5	44·5	49·5	54·5	63·5	76·0	94·0	·0022	·0011	·010	·0016
50	—	—	7·0	10·5	14·0	21·0	28·0	35·0	42·0	49·0	56·0	63·0	69·0	81·0	98·0	123	·0018	·0009	·0081	·001[illegible]
75	—	—	11·0	16·5	22·0	33·0	44·0	54·0	64·0	74·0	84·0	94·0	104	123	150	193	·0014	·0007	·0063	·001[illegible]
100	—	—	14·0	22·0	28·0	42·0	56·0	70·0	84·0	98·0	112	126	139	165	203	263	·0010	·0005	·0045	·000[illegible]
125	—	—	18·0	27·0	36·0	54·0	71·0	88·0	105	122	139	156	173	206	254	331	·0008	·0004	·0036	·0006
150	—	—	21·0	31·5	42·0	63·0	84·0	105	126	146	166	186	206	246	303	398	·0006	·0003	·0027	·000[illegible]

In sizing sub-atmospheric systems, mains should be carried 2 in. dia. to the end minimum.

Condense Pipes—Sizing of the condense piping can only be empirical, since return lines commonly run partially filled with water, the remainder being filled with vapour and air. Part B of the table gives suitable sizes for various conditions.

Noise—In heating systems where absence of noise is important an upper limit is set to the flow in steam risers by the velocity at which globules of condensation are carried up with the steam. Approximate maximum dis-

charges for upward risers are indicated by the dotted line on Table LVIII, A.

Sizing by Velocity—Piping in boiler houses, such as main headers, exhaust steam manifolds, etc., are often sized on velocity only and not on a pressure drop basis. The volume of steam at various pressures may be taken from Table LX (p. 351), and the sizes based on velocities such as those given in Table LIX.

Then

$$\text{Pipe area (sq. ft.)} = \frac{\text{Wt. of steam (lbs. per hour)} \times \text{Vol. of 1 lb. steam}}{\text{Velocity (ft. per min.)} \times 60}.$$

TABLE LIX

STEAM VELOCITIES (SATURATED STEAM)

	Ft. per Minute
Boiler outlet connection (L.P.)	900–1500
Boiler outlet connection (H.P.)	3000–4000
Main H.P. and L.P. steam headers	4000–6000
H.P. steam lines	6000–9000
Calorifier connections	5000–6000
Exhaust steam piping	4000–6000
Exhaust piping under vacuum	20,000

CROSS-SECTIONAL AREA OF PIPING

Nom. Dia., In.	Area, Sq. In.	Area, Sq. Ft.
2	3·14	·0218
2½	4·92	·0341
3	7·07	·0491
4	12·57	·0873
5	19·63	·1364
6	28·27	·1964
7	38·48	·2673
8	50·27	·3491
9	63·62	·4418
10	78·54	·5454
12	113·10	·7854

Pumping Return Mains—The discharge pipes from condense return or vacuum pumps run full of water, and may therefore be sized as for hot-water pipes from Table XXXIX, p. 196.

EXPANSION

When steam is turned on to a cold system of piping expansion takes place. Hangars previously vertical become inclined; where passing through a wall the paint work will reveal the extent of the expansion. The forces of expansion are immense and will bend or fracture any branch, joint or fixing tending to restrain movement.

When steam is turned off and the system cools down, the piping contracts and should return to its original place. It will do so if it is anchored at suitable points, but if not it may creep along by successive cycles of heating and cooling until failure occurs at the weakest point.

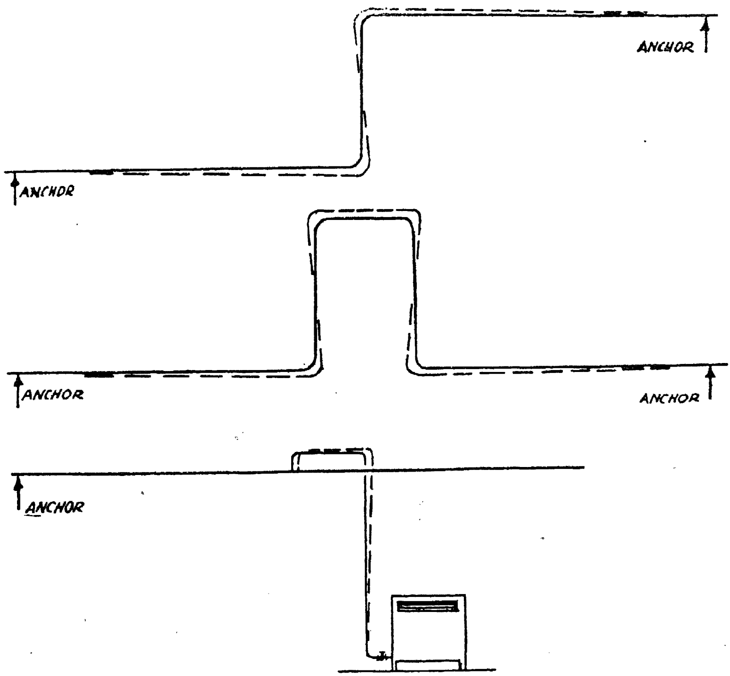
The amount of expansion may be calculated by multiplying the coefficient of linear expansion for the material used, by the temperature rise, and by the length of pipe. Table LIXA gives this, per 100 ft. of pipe, for various steam pressures, assuming the system is erected cold at 40° F. The expansion of condense piping, being at a lower temperature due to cooling, is usually taken at $\frac{2}{3}$ that of the steam, if of steel, and the same as for steel steam piping if the condense pipe is of copper.

TABLE LIXA. EXPANSION OF STEEL STEAM PIPING

Gauge Pressure	Temp. °F.	Expansion from 40° F. per 100 Ft. in Inches	Gauge Pressure	Temp. °F.	Expansion from 40° F. per 100 Ft. in Inches
1	215	1·41 in.	30	274	1·89 in.
3	221	1·47 „	50	298	2·08 „
5	227	1·52 „	75	320	2·26 „
10	240	1·61 „	100	338	2·40 „
15	250	1·69 „	125	353	2·52 „
20	259	1·76 „	150	366	2·62 „

The movement due to expansion requires to be carefully considered and allowed for in the lay-out.

The basic principle in using the *piping* itself to take up the expansion, as opposed to taking it up by special *joints*, is that bends must be introduced into the piping in one way or another, i.e.



Consider a straight run of 8 in. steel pipe 200 ft. long fixed at its ends, unstrained initially and heated to 280° above the air temperature. The expansion resulting (Table LIXA) will be $2 \times 2{\cdot}26 = 4{\cdot}52$ in. It would require a force of about 160 tons to compress this pipe to its original length, and the pipe would fail in direct compression even assuming that it was held to prevent buckling. In practice it would generally buckle and so relieve the stress.

Now if a well-designed expansion loop were introduced into the run, it can be shown that the force exerted at the anchor points will be of the order of 3 tons or less.

If the runs are short (say 50 ft. or under) the normal changes of direction are usually adequate, provided the piping is hung so as to be free to move in any direction. Branch connections should be taken off with bends so as to leave the run-outs flexible, otherwise leakage of joints will result. Connections to radiators and other apparatus should be free of all strain.

Where long straight runs of steam and condense piping occur, use may be made of one of the following types of expansion device.

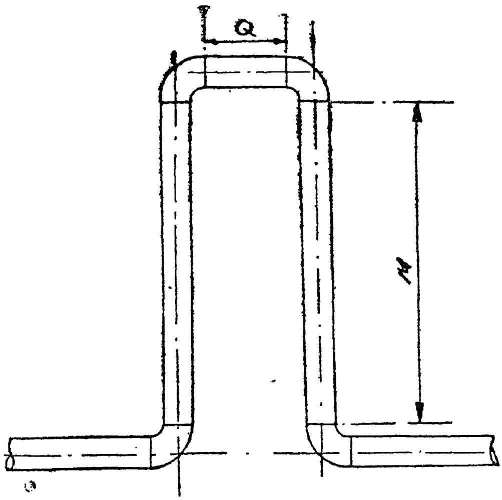
(*a*) Sliding expansion joints (see Fig. 139A, p. 220, as used for hot water). These are used mainly in trenches or ducts where other methods are impossible. They are liable to give trouble with gland leakage. The lubricant must be one which does not melt or seize up solid at steam temperatures. They require accurate guiding of the main to prevent angular movement, and careful maintenance.

(*b*) 'Concertina' or 'Bellows' type joints. These usually give small movement and impose great strains on the piping. They are mostly of American or Continental manufacture.

(*c*) 'Horseshoe' or 'Lyre' type expansion loops (see Fig. 139B for hot water). These may be placed vertically with a drip point at inlet side, or horizontally, and are generally the best where space can be provided. The amount of expansion per loop is given in the above-mentioned figure. A variation of this is the 'Bugle' type, which is in effect one loop of a vertical spiral. All types of loops are made by bending a straight length of pipe to the required shape, except for the larger sizes where they are usually in three sections with welded or flanged joints. These loops are sometimes made of copper, but they then have to be taken down for annealing from time to time.

(*d*) 'Made-up' loops formed with four bends and straight lengths of piping welded between. If the legs are of adequate length, a greater amount of expansion may be catered for by this means

than with any of the other types, so that fewer loops are required in a given run. The forces involved are generally less with this system than with (*b*) and than with many types of (*c*).



NOMINAL PIPE SIZE	EXPANSION INCLUDING COLD DRAW	DIMENSION 'M.'	DIMENSION Q.
2"	3½"	4'-6"	1'-6"
3"	3½"	5'-0"	2'-0"
4"	3½"	6'-6"	3'-0"
5"	3½"	7'-6"	3'-0"
6"	5"	11'-6"	3'-6"
8"	5½"	14'-0"	4'-0"
10"	5½"	19'-0"	4'-0"
12"	6"	25'-0	4'-0"

FIG. 194A.

Typical dimensions are given in the figure 194A.

The dimensions are based on elasticities and allowable stresses applicable to the pipes at temperatures up to about 460° F. At high temperatures the properties of mild steel are profoundly modified, but this is outside the range of heating practice.

It will be noted that in this figure short radius bends are shown, welded to the straight lengths. These are known in the United States as 'Tubeturns' or 'Welding Ells', and their use in this country will no doubt increase. They save much space as compared with long radius bends, and though their pressure loss is higher, this is unimportant in most steam

systems. Tees, reducers and other fittings are also available made out of one piece similar to the bends.

Cold-Draw—In order to reduce the forces in the pipe when heated, expansion loops should be sprung open when being fixed cold (known as cold-draw), by an amount equal to 40 or 50 per cent. of the total movement at each loop. On heating up the tension will then pass to compression through a neutral point at mid-temperature, and at the working temperature the forces will be roughly halved, as compared with a system having no cold-draw.

Anchor Points—An Anchor Point is a clamp bolted or welded to the pipe, and secured to a rigid base or fixture such that the pipe is held immovable at that point under all conditions of stress set up by expansion and contraction. On a long run the main should be divided into a number of sections of 100 to 300 ft. according to the amount of movement to be allowed for. Each section should be anchored at the ends with an expansion loop near the centre, or the centre of the loops may be anchored and the piping left free between. If a sliding expansion joint is used, this may also form the anchor point by solidly bolting the body to a fixed support.

Pipe Supports—Where it is possible to use hangers for the piping, these are preferable to other forms of support, as they allow freedom of movement in all directions. They may be slung from roof trusses, wall brackets or from floor standards of inverted U or 'jibbet' form.

Plate XX shows external steam and condense mains supported on a lattice girder spanning some 70 ft. across rails, road, etc., at a large depot carried out to the designs of the Authors.

Where pipes are in ducts or trenches, rollers and chairs are usually adopted. In such case, guides to prevent lateral movement are necessary at intervals. Standard supports may be seen in manufacturers' catalogues.

Where sections of a run rise vertically for any considerable distance, as where external mains run over a road or railway, the upper portions, in addition to themselves expanding, will be lifted bodily by the expansion of the vertical legs. This lift will take the weight off the hangers of the high-level portion, and leave unduly long spans virtually unsupported, with risk of overstressing the pipe. To obviate this, one of the various types of spring-loaded hangers may be used. These are so arranged that as the pipe lifts, the hanger continues to support it adequately.

Steam Main Drainage (Fig. 194B)—Where possible, steam mains should fall in the direction of steam flow at about 1 in. in 20 ft. Reducers should be eccentric to allow no pocket for condense to collect. At the end of a run the main should be drained with a steam trap discharging to condense main or to waste if no return exists. Mains longer than about 300 ft. should be drained intermediately.

Where it is necessary to step the steam main up to a higher level due to the fall having brought the pipe too low, this may be done by arranging a 'relay' suitably drained as shown. This may be combined with provision

for expansion at the same point by taking the vertical pipe up in the form of a loop.

The size of trap must be adequate to handle the initial condensation rate rapidly, and the size should be calculated on at least three times the normal rate.

Where the steam main cannot fall in direction of flow, it should fall in the opposite direction and its size increased to reduce the velocity. A drainage point should be provided at the lowest point.

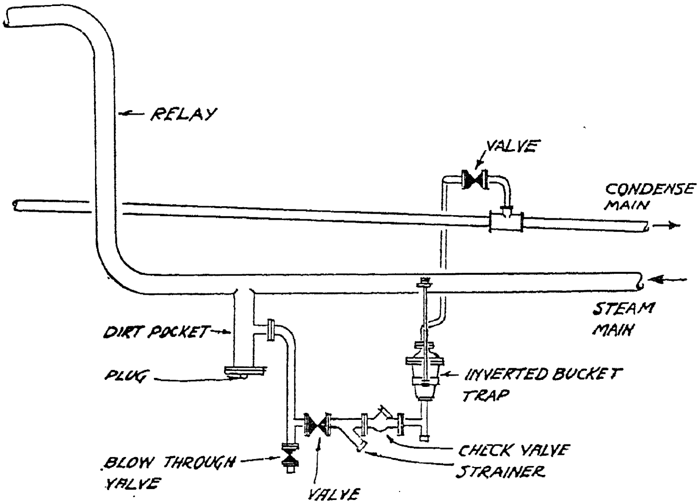


FIG. 194B.—Steam Main Drip Point and Relay. Flanged joints shown for high-pressure screwed joints would be used for medium and low pressures.

It is best not to drain steam mains through heating apparatus, such as unit heaters, radiators, calorifiers, etc., but to take connections to these from the top or sides of the mains so that they receive dry steam. This does not apply in the case of short run-outs which will satisfactorily drain themselves through the apparatus.

STEAM ACCESSORIES

Boiler Feeder—The boiler feeder indicated in Figs. 186, 187 and 188 (*a*), sometimes referred to as a 'Banjo', is given in detail in Fig. 195. It consists,

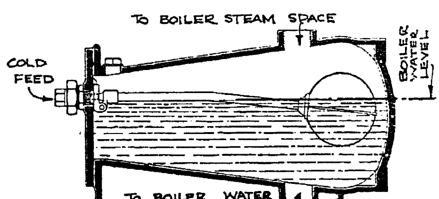


FIG. 195.—Boiler Feeder.

in effect, of a ball valve contained in a casing connected top and bottom direct to the boiler so that the water levels in both are the same. As the level drops, the valve opens and water from the feed tank is admitted. The tank must, of course, be high enough for the water to overcome the pressure in the boiler. For instance, if the steam is at 5 lbs. per sq. inch pressure, the tank requires to be $5 \times 2{\cdot}32 = 11{\cdot}6$ ft. above water line; an allowance of 20 ft. would be suitable.

The tank need only be of a nominal capacity such as 20 gallons for a small system to 50 gallons for a large system, since this method of feed is not used where any appreciable quantity of make-up is required.

Radiator Valves—Ordinary screw down valves are commonly used for controlling steam radiators and convectors, etc. Where it is desired to prevent inward air leaks as with a sub-atmospheric system a glandless type of valve is to be preferred.

Fig. 196 (*a*) shows a type of radiator valve having no gland, and known as the packless type. The seal between steam and air is formed by a flexible copper bellows.

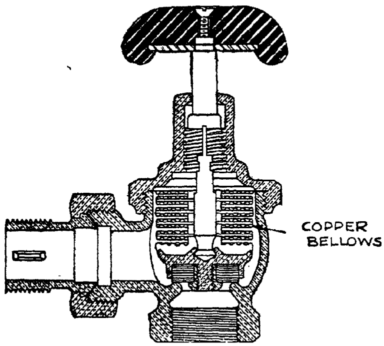


FIG. 196 (*a*).—Packless Radiator Valve (Dunham).

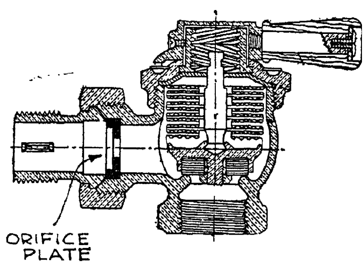


FIG. 196 (*b*).—Orifice Plate fitted to Packless Valve (Dunham).

Fig. 196 (*b*) shows a similar type of valve with an orifice disc which allows its being used for modulating vapour and sub-atmospheric systems.

Thermostatic Trap—The purpose of this and other steam traps is to allow condensation to pass but not steam.

One type of thermostatic trap is shown in Fig. 197 (*a*). It consists of a flexible copper bellows containing a volatile spirit. When the trap is cold

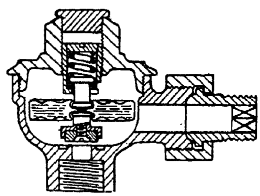


FIG. 197 (*a*).—Thermostatic Trap.

the port is open. When the steam arrives after passing through the radiator or other piece of apparatus, the spirit is volatilized and the valve of the trap is shut. Condensation then begins to collect and eventually allows the trap to cool down and re-open.

Bucket Traps—For the draining of condensation from larger items of plant such as calorifiers, air heaters, unit heaters, steam mains, etc., the bucket type of trap is usually adopted.

There is a wide variety of designs to suit different applications and pressures, one of which is shown in Fig. 197 (*b*).

When condensation is entering the trap the inverted bucket is at the bottom and the valve open. As steam begins to enter, the bucket rises to the position shown in the figure, closing the valve against further discharge. As the steam condenses the bucket drops so as to allow further condensation to pass.

When discharging to a condense main at a higher level than the trap, a non-return or 'check' valve is fitted to the outlet of the trap. The condense is then lifted by the steam pressure behind it, but is prevented from returning by the check. (See 'Lifting Traps' later.)

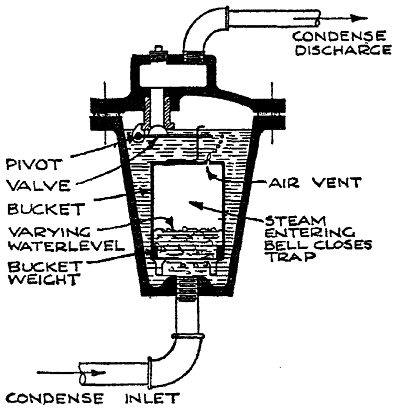


FIG. 197 (*b*).—Bucket Trap (Drayton Armstrong).

The capacity of discharge of bucket and thermostatic traps depends on the steam pressure and on the design. Makers' lists should be referred to for the necessary data.

Float Trap (Fig. 197 (*c*))—This type of trap is preferred by some to the bucket type, as it gives a continuous discharge. It is usually limited in regard to pressures.

Float and Thermostatic Trap (Fig. 197 (*d*))—For low pressure and vacuum steam this trap is most suitable for heavy duties above the range of the ordinary thermostatic type. The float valve allows water to pass freely, and the thermostatic element releases the air which, particularly under vacuum, is large in volume.

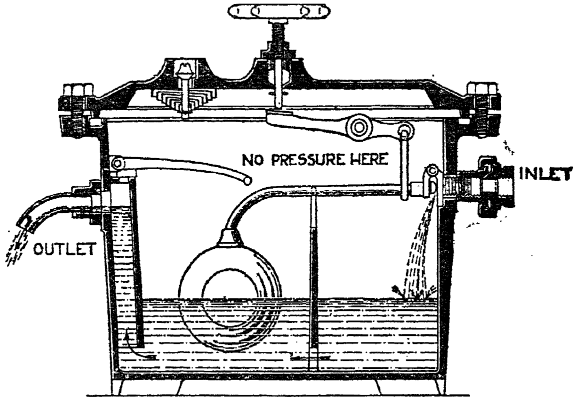


FIG. 197 (*c*).—Float Trap (Royles).

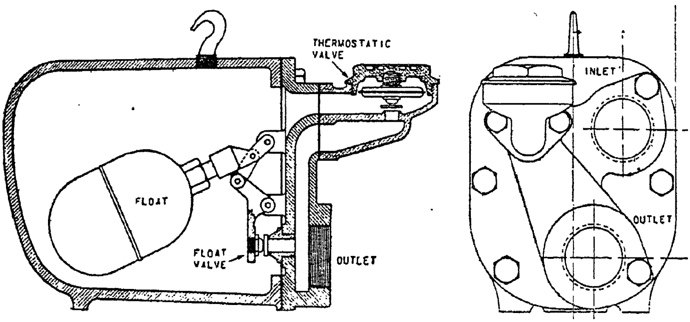


FIG. 197 (*d*).—Float and Thermostatic Trap (Dunham).

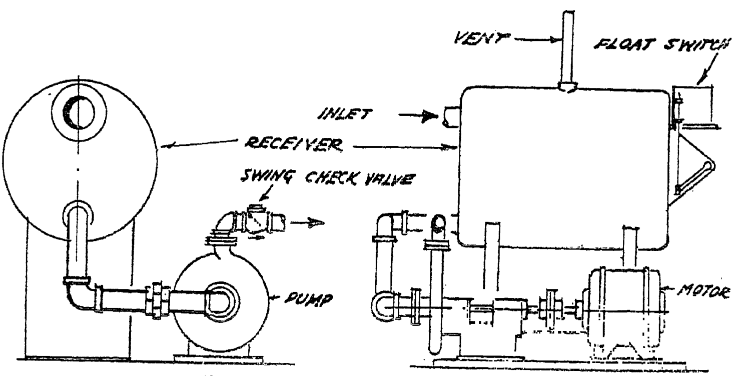


FIG. 197 (*e*).—Condense Return Unit.

Condense Return Pumping Unit (Fig. 197 (*e*))—This is used to collect and return water of condensation when, due to level or distance, it cannot

be returned by gravity, and where the vacuum system is not used. It consists of a receiver containing a float switch, and a centrifugal pump, motor-driven, stopped and started by the float switch. The rating of the pump is usually about three times the condense return rate. The receiver requires a large vent to deal with the vapour from re-evaporation. It is necessary for the pump to be lower than the receiver to provide a 'head' on the pump suction.

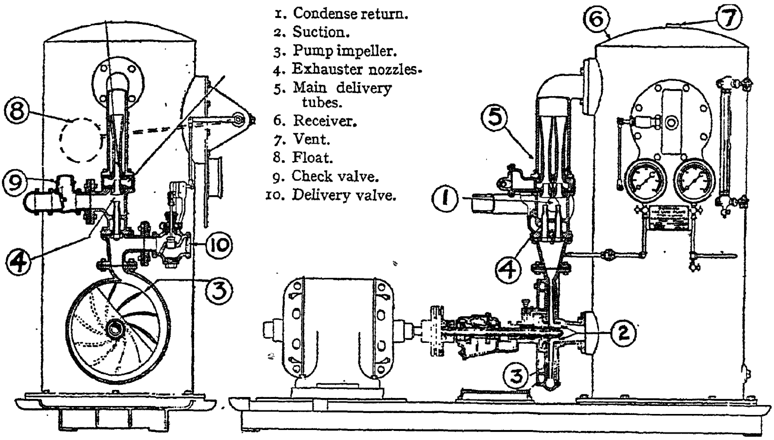
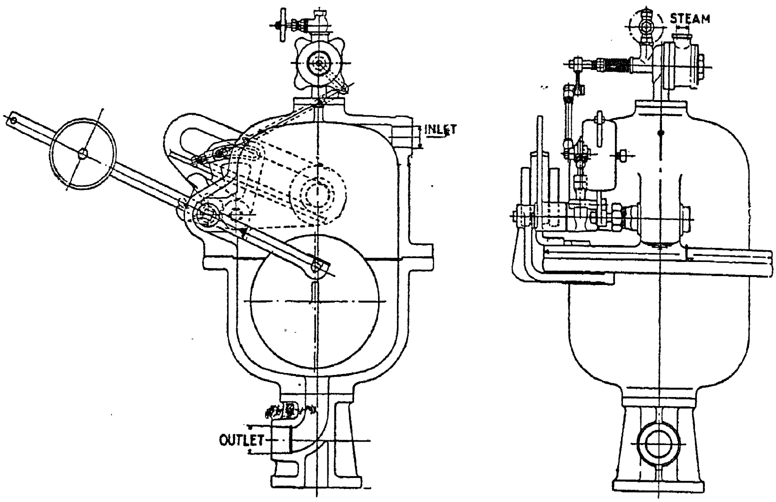


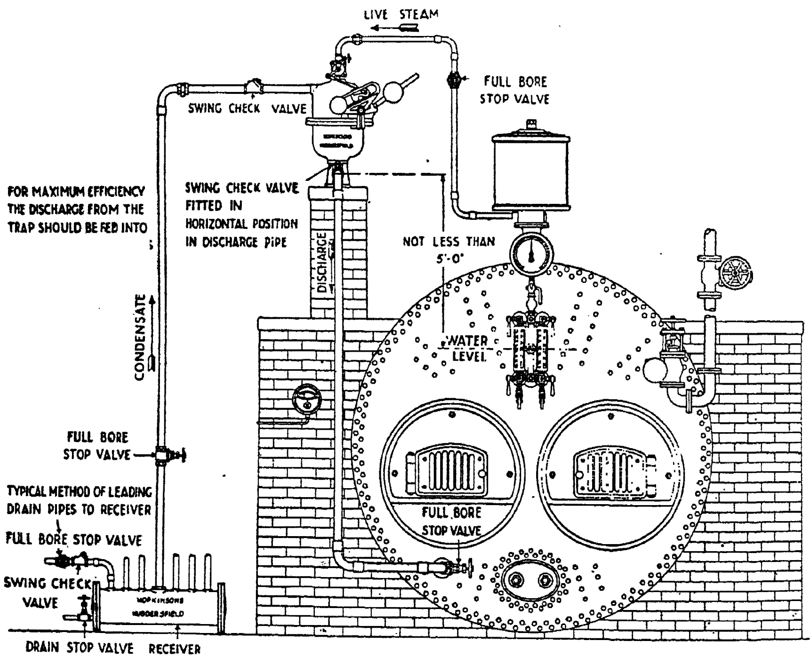
FIG. 197 (*f*).—Vacuum Pump (Dunham).

Vacuum Pumping Unit—Fig. 197 (*f*) illustrates one form of electrically-driven vacuum pumping set. The receiver is the same as above, and stops and starts the pump according to the level of water. The pump discharges through two outlets; one through a controlling valve to the pipe return to hot well or boiler, and one through a set of nozzles into the air-separating cylinder. The action of the nozzles is to eject air and vapour from the system, thus producing the required vacuum. A vacuum switch stops and starts the pump independently of the receiver float switch so as to maintain a constant negative pressure, but the receiver float switch over-rides this, so as to remove the condense as it collects. The air-separating cylinder has a float operating the control valve in the discharge to hot well or boiler, so as to keep a constant level. The returns first pass through a strainer, as shown. The air separator is at atmospheric pressure, having a vent from the top.

Alternating Trap (Fig. 197 (*g*))—The action of this has been described under the vapour system. It is also of use in returning condensate on a closed system with a boiler at any pressure, but is usually confined to the smaller systems. In this case condense collects in the casing, which is at atmospheric pressure, being open through the vent. When the float rises



Detail of Trap.



General Arrangement showing Trap connected to Boiler and Receiver.

FIG. 197 (*g*).—Alternating Trap (Hopkinson).

to the top the arrangement is such that a lever is thrown over, shutting the vent and opening the steam connection from the boiler. The pressures are thus balanced and the condensate can return through a check valve to the boiler by gravity, due to the elevated position relative to the boiler water-line. When this occurs the check valve on the inlet is held shut. A small receiver is necessary to hold the condense during this period. When the trap is empty the valves are returned to the original position and the cycle repeats.

Lifting Traps (Fig. 197 (*h*))—Steam traps may be made to lift the condensate to an overhead return pipe provided there is adequate steam pressure to overcome the head of water. 1 lb. per sq. in. will support a column of water 28 ins. high. There is the resistance of trap, pipe and fittings to

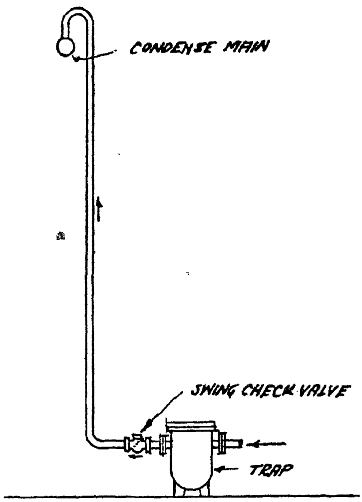


FIG. 197 (*h*).—Lifting Trap.

overcome, and it is usual to assume that if steam is at x lbs. per sq. in. pressure the height it is possible to lift will be $2x$ ft. The trap must be of a type in which the outlet and not the inlet is controlled by the float or thermostatic device. The check valve is usually fitted on the discharge, but with inverted bucket traps it is better on the inlet, to prevent re-evaporation of the water forming the 'seal' of the trap should the steam pressure be variable.

Super-Lifting Traps (Fig. 197 (*i*))—Where the condense is from steam at too low a steam pressure to lift the condense the required height, use may be made of the 'super-lifting' trap. The operation is similar to an alternating trap, described above, which may also be used for this duty. The steam inlet is connected to a high-pressure steam main. At least 30 lbs. per sq. in. is necessary. A small vented receiver is necessary to accommodate the condensate during period of discharge. Where there is no steam

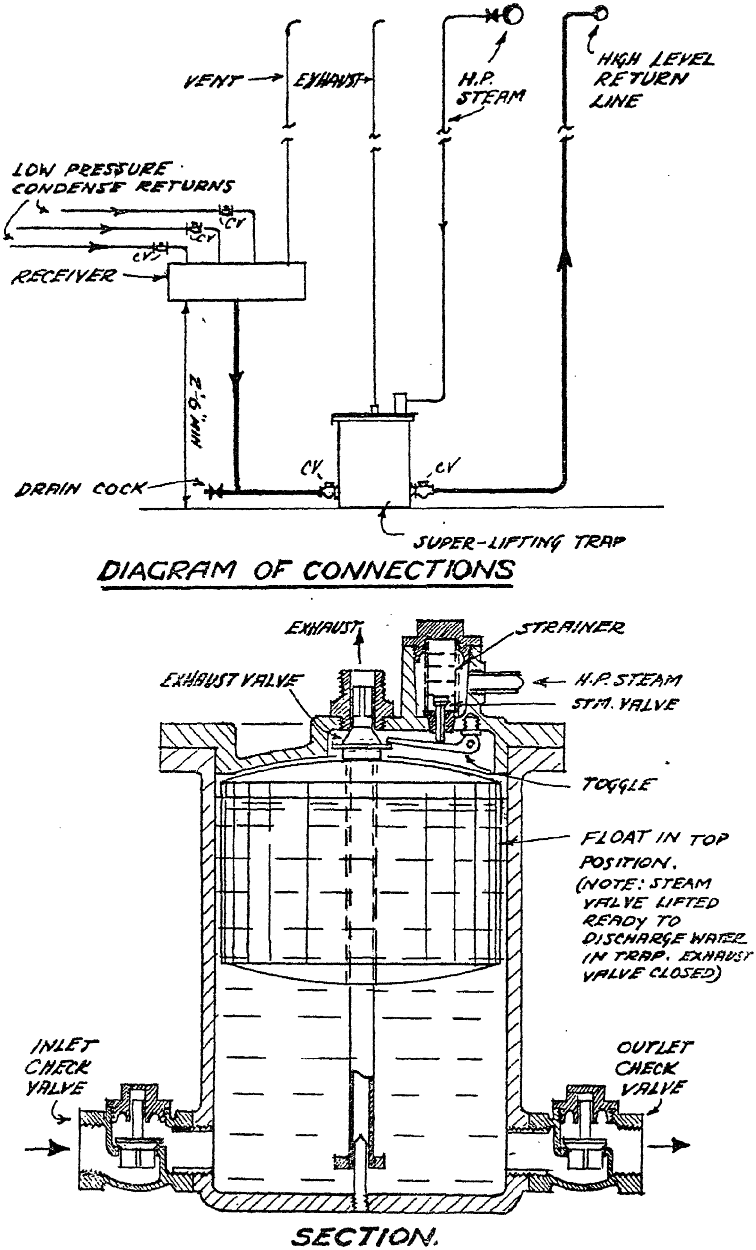


FIG. 197 (i).—Super-Lifting or Pump Trap.

at suitable pressure, a condense pumping unit will be required, or if the load is small the condense must be run to waste.

Reducing Valve—Fig. 198 (*a*) shows one of the many forms of direct-acting reducing valve. The cylinder at the base contains an extensible rubber sleeve reinforced with rings outside. The outside springs obviously tend to hold the valve open, but as steam pressure enters and builds up on the outlet side, the valve tends to be forced downwards on to its seat,

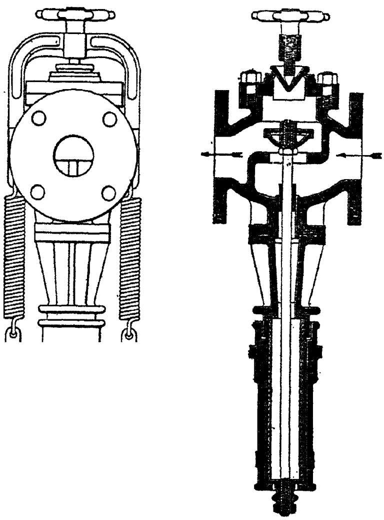


FIG. 198 (*a*).—Steam Reducing Valve (Royles).

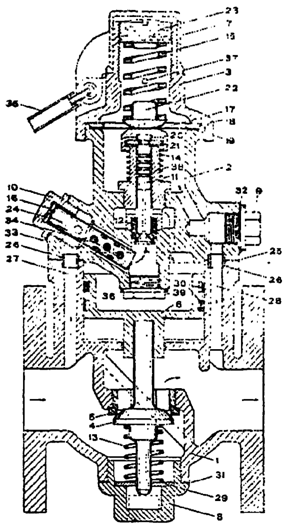


FIG. 198 (*b*).—Reducing Valve, Relay Type (Hopkinson).

so throttling the steam and reducing its pressure. In practice a state is achieved such that the valve establishes itself at a point where the pressure of the springs upward and the outlet pressure downward are balanced and the reduced steam pressure is maintained steady. It will be noted that a handwheel is provided at the top of the reducing valve which permits of adjustment of the tension of the springs (hence of the reduced pressure) and at the same time of the safety valve setting. The top inverted cone, it will be seen, serves as a safety valve.

Fig. 198 (*b*) shows one form of '*Relay*' operated reducing valve. This type gives more uniform outlet pressure with a variable inlet pressure. In this case the main valve (4) is operated by the piston (6), steam being admitted to the top side of the piston (36) from the high pressure side via the pilot valve (11). The pilot valve rises or falls with variations of the low-pressure steam which communicates through passage (28) with the diaphragm (17). The reduced pressure is adjusted by varying the tension in the spring (15). A strainer (34) filters the high-pressure steam admitted

to the pilot valve. In practice a balance is set up with the main valve (4) opened just sufficiently to maintain the correct pressure on the reduced side.

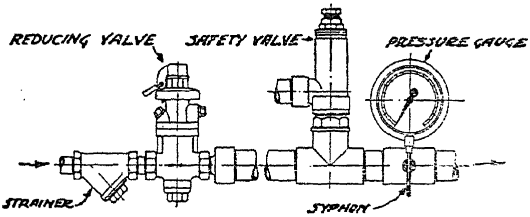


FIG. 198 (c).—Arrangement of Reducing Set.

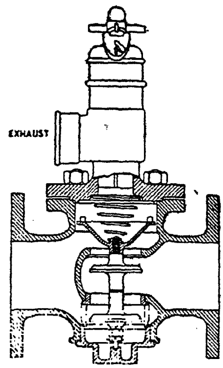


FIG. 198 (d).—Excess Pressure Isolating Valve (Dewrance).

The Factories Act 1937 requires that where a reducing valve is fitted there shall be provided:

Sec. 30 (*b*) 'A suitable safety valve so adjusted as to permit the steam to escape as soon as the safe working pressure is exceeded, or a suitable appliance for cutting off automatically the supply of steam as soon as the safe working pressure is exceeded.'

A pressure gauge and stop valve is also required.

To comply with the above requirements the safety valve or valves must be capable of discharging the full volume of steam passed by the reducing valve when full open, and this calls for large valves. Fig. 198 (*c*) shows the arrangement.

The alternative mentioned above of device to cut off the steam as soon as the working pressure is exceeded is termed an 'excess pressure isolating valve', one type of which is shown in Fig. 198 (*d*). On excess pressure occurring the small pilot safety valve begins to blow, and this upsets the balance between the top and bottom of the piston, which then tends to rise. When the pressure has fully built up on the underside of the piston the steam is completely shut off.

Steam Equipment Generally—In addition to the above brief list of steam apparatus particularly concerned with heating systems, there are numerous other fittings and equipment to which it is not possible to refer in detail.

These include:

Feed water treatment plant.
Feed pumps and injectors.
Hot well arrangements and fittings.
Boiler mountings for high pressures.
Steam valves of various types.
Condense valves.
Steam separators.

Flanged joints.
Welding and screwing.
Cast iron and cast steel fittings.
Steam and feed water meters.
Instruments.
Controls.

For further information on these matters, reference should be made to a textbook on Steam Engineering, and to the various makers' catalogues and handbooks.

APPENDIX

PROPERTIES OF STEAM

Saturated Steam is steam in contact with the water from which it has been evaporated.

Dry Saturated Steam is saturated steam with no entrained moisture.

Wet Steam is steam containing small globules of moisture carried over from the boiler or caused by partial condensation.

Superheated Steam is steam heated out of contact with the water from which it was evaporated.

Absolute Pressure—The absolute zero of pressure is that state which exists in a perfect vacuum. Absolute pressures are given in lbs. per sq. in. above this zero.

Atmospheric Pressure referred to the absolute scale is 14·6959 (approx. 14·7) lbs. per sq. in. at standard barometric pressure of 30·01 ins. of mercury at 62° F.

Gauge Pressures are pressures commencing from the datum of atmospheric pressure as read from a gauge, the outside of which is subject to the atmosphere. Since atmospheric pressure is variable according to the barometric reading, gauge pressures are to this extent also variable. Standard gauge pressures are referred to standard atmospheric pressure.

A gauge pressure of, for example, 50 lbs. per sq. in., is the same as 64·7 lbs. per sq. in. absolute at standard atmospheric pressure.

Sub-Atmospheric Pressures are generally referred to in inches of mercury (symbol Hg) of vacuum below atmospheric pressure. Thus 25 ins. Hg vacuum is equivalent to an absolute pressure of 5 ins. Hg column or 2·45 lbs. per sq. in. (weight of 1 cu. in. of mercury 0·491 lbs.).

Temperature—The temperature of saturated steam is the same as that of the water from which it has been generated, and with which it is in contact.

The boiling point of water at standard atmospheric pressure is 212° F., and saturated steam at that pressure will have the same temperature. If the steam is enclosed and the pressure made to rise by the addition of more

heat, the temperature will rise above 212°, e.g. at 50 lbs. gauge, temperature = 297·6° F. Similarly if heat is removed from steam at atmospheric pressure in a closed vessel, as by condensing it, its pressure will drop, and the temperature likewise. Thus, at 25 in. Hg vacuum, the temperature of steam is 133° F. Fig. 199 gives a curve showing how the temperature varies with pressure over the range covered by heating practice.

Heat of Steam—*Latent heat* is the heat to be added to water at its boiling point to evaporate the water into steam at the same temperature. At atmospheric pressure the latent heat is 970·6 B.T.U.'s per lb. Increase of pressure reduces the latent heat; decrease of pressure raises it, as will be seen from Fig. 199.

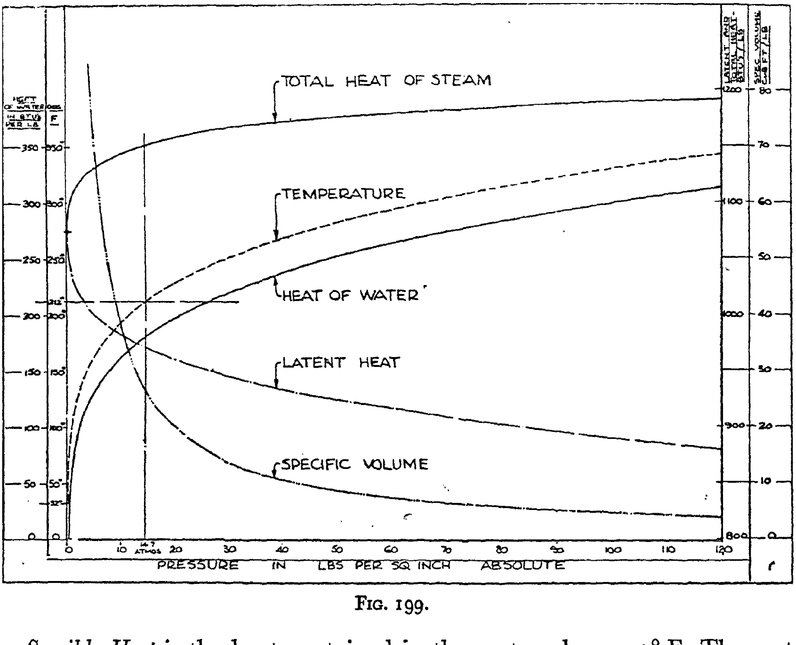


FIG. 199.

Sensible Heat is the heat contained in the water above 32° F. Thus, at 212° F. taking specific heat of water as 1,* the sensible heat is 180 B.T.U.'s per lb. The curve thus follows directly the temperature curve.

Total Heat is the sum of the latent and sensible heat. At atmospheric pressure it is thus 970·6 + 180 = 1150·6* B.T.U.'s per lb. It will be noted from Fig. 199 that with increasing pressure the total heat rises even though the latent heat has fallen, due to the sensible heat having more than made up the balance. The total heat reaches a limit at 400 to 450 lbs. gauge, after which it begins to fall.

Weight of Steam—Steam is usually referred to in terms of pounds weight. If the heat required for a certain heating system is in B.T.U.'s, the weight of

* Actually mean specific heat not exactly unity. Total heat at atmospheric pressure = 1150·7.

steam to be condensed to supply that heat is arrived at by dividing the B.T.U.'s by the latent heat at the pressure of the steam supplied. If in the process of exchanging heat some of the sensible heat is recovered, then the B.T.U.'s are divided by the total heat minus the difference between the heat of the outgoing condensate and 32°.

Naturally the weight of outgoing condensate will equal the weight of the steam inlet, though some of the condensate may re-evaporate, as explained already.

Volume of Steam—The volume of 1 lb. of water at 212° is ·0167 cu. ft. When converted into steam at atmospheric pressure the volume is 26·8 cu. ft.

As the pressure is increased the specific volume decreases, e.g. at 50 lbs. per sq. in. the volume is 6·68 cu. ft./lb.

As the pressure decreases below atmospheric, the specific volume increases rapidly, e.g. at 25 in. Hg vacuum it is 143·9 cu. ft./lb.

Superheating of Steam—If steam is heated out of contact with its water its temperature is raised and the sensible heat increased, but the latent heat remains the same.

If dry steam at one pressure is expanded to a lower pressure without doing work, superheating also occurs. The total heats before and after expansion remain the same, but the total heat of saturated steam at the lower pressure would be less, so that the difference goes to superheat the steam. The number of degrees of superheat can be calculated from the steam table LX as follows:

$$\text{Superheat degrees} = \frac{\begin{array}{c}\text{Total Heat Initial} - \text{Total}\\ \text{heat at final pressure saturated}\end{array}}{\begin{array}{c}\text{Mean specific heat of superheated}\\ \text{steam at final pressure}\end{array}}.$$

In the case of wet steam, some of the surplus heat will be taken up in evaporating the moisture. The *wetness fraction* is the ratio of moisture present to total weight.

The *specific heat* of superheated steam at atmospheric pressure may be taken as ·47 B.T.U.'s per lb. at constant pressure. This rises at higher pressures and temperatures.

Entropy—This is a conception which is much used when considering steam used in a heat engine. For ordinary heating problems it is not so important, and need only be considered very briefly.

Entropy does not correspond to any physical property of heat of which we can have direct knowledge. It is simply a mathematical ratio for expressing the availability of heat, and the entropy of any *closed* system involving heat-transfer increases as the transfer takes place, e.g. from steam at 300° to secondary water at 50° in a calorifier, where finally we are left with condensate at, say, 200° and water at 180°. By no known method could the heat be taken out of the water and put back into the

condensate to make steam, and this irreversible process takes place in the direction of increase of entropy. Similarly, the expansion of steam through a throttling valve is an irreversible process which causes an increase in entropy of the steam.

Entropy may be depicted graphically by drawing a curve of the heat content of a closed system, the curve having ordinates of absolute temperature. If the area under the curve represents heat, then the abscissae represent entropy.

For the development of this subject the reader is referred to any of the standard works on Heat Engines.

Steam Boiler Evaporation—Reference to makers' catalogues will show that cast iron and steel low pressure boilers are listed in B.T.U.'s per hour, it being assumed that all condense is returned on a closed circuit. Larger boilers suitable for higher pressures are commonly listed in 'pounds of steam per hour, *from and at 212° F.*'

This denotes that if the water in the boiler is at atmospheric pressure the steam generated at that pressure will be the quantity given, i.e. it is assumed that only latent heat is added.

Such a rating, however, is of little practical use as it stands. The feed water is seldom exactly 212° and the pressure is usually above atmospheric.

For any other set of conditions the Actual Evaporation can be derived as follows:

$$\text{Actual evaporation} = \frac{\text{Evaporation from and at } 212^\circ\text{ F.} \times 970{\cdot}6}{\left.\begin{matrix}\text{Total heat at pressure}\\ \text{required}\end{matrix}\right\} - \left\{\begin{matrix}\text{Difference between}\\ \text{feed water temp. and } 32^\circ\end{matrix}\right.}.$$

Factors of evaporation will be found in numerous data books for various pressures and feed temperatures, but they can be worked out from first principles in the manner stated.

It will generally be found that the Actual Evaporation is less than the 'from and at' rating, which means, of course, a larger boiler for a given duty.

Table of Properties of Saturated Steam—Table LX gives the properties of saturated steam over the range required for most heating problems. The figures have been calculated for exact whole numbers of gauge pressure, this being the most useful form for use in heating.

TABLE LX

PROPERTIES OF DRY SATURATED STEAM

	INCHES H.G.	ABSOLUTE PRESSURE LBS/SQ.IN.	TEMPERATURE °F	HEAT B.T.U./LB. IN THE WATER	LATENT HEAT	TOTAL HEAT	VOLUME STEAM CU.FT./LB.
VACUUM	25	2·4	133·6	101·5	1018·1	1119·6	143·9
	22·5	3·7	149·5	117·4	1008·8	1126·2	98·4
	20	4·9	161·4	129·3	1001·7	1131·0	75·1
	18	5·9	169·2	137·1	997·1	1134·2	63·3
	16	6·9	175·9	143·8	993·1	1136·9	54·8
	14	7·8	181·9	149·8	989·4	1139·2	48·4
	12	8·8	187·3	155·3	986·1	1141·4	43·3
	10	9·8	192·2	160·2	983·2	1143·4	39·2
	8	10·8	196·8	164·8	980·4	1145·2	35·9
	6	11·8	201·0	169·1	977·7	1146·8	33·1
	4	12·7	204·9	172·9	975·3	1148·2	30·7
	2	13·7	208·6	176·6	972·9	1149·5	28·6
GAUGE PRESSURE	LBS/SQ.IN. 0	14·7	212·0	180·1	970·6	1150·7	26·8
	2	16·7	218·5	186·7	966·4	1153·1	23·8
	4	18·7	224·4	192·7	962·6	1155·3	21·7
	6	20·7	229·8	198·1	959·2	1157·3	19·5
	8	22·7	234·8	203·2	955·9	1159·1	18·0
	10	24·7	239·4	207·8	952·9	1160·7	16·5
	15	29·7	249·8	218·4	946·2	1164·6	13·9
	20	34·7	258·8	227·6	940·0	1167·6	12·0
	25	39·7	266·8	235·7	934·6	1170·3	10·6
	30	44·7	274·0	243·0	929·5	1172·5	9·47
	40	54·7	286·7	256·1	920·4	1176·5	7·83
	50	64·7	297·6	267·4	912·2	1179·6	6·68
	60	74·7	307·3	277·2	905·2	1182·4	5·83
	70	84·7	316·0	286·2	898·8	1185·0	5·19
	80	94·7	324·0	294·6	892·5	1187·1	4·66
	90	104·7	331·2	302·0	887·1	1189·1	4·24
	100	114·7	338·0	309·1	881·6	1190·7	3·89
	125	139·7	353	324·6	869·2	1193·8	3·23
	150	164·7	366	338·7	857·9	1196·6	2·75
	200	214·7	388	362·2	838·2	1200·4	2·13
	250	264·7	406	381·7	821·2	1202·9	1·74

INTERPOLATED FROM TABLES

CHAPTER XV

Heating by High-Pressure Hot Water

The two main heating media so far discussed, i.e. steam and hot water at normal pressure, each suffer from certain disadvantages, which may be summarised thus:

Steam. Mains require careful draining, restricting frequent change of levels. All such drain-points, and all steam-using apparatus, require steam-traps, check-valves, etc., which are costly to install and to maintain.

It is not generally possible to recover all the heat contained in the steam. This point is discussed in detail on p. 354.

If the condensed steam is returned to the boilerhouse, the condense-main is a source of trouble and expense due to corrosion.

Low-Pressure Hot Water. The heat-carrying capacity of the medium is limited by its low temperature and the small permissible temperature drop. Heating apparatus and mains are therefore unduly large and expensive. The system is unsuitable for heating large spaces by unit heaters, which are relatively ineffective with hot water at less than, say, 190° F.

To meet these and other disadvantages, there has been evolved a system of heating, using as a medium water at a high temperature (about 300° F. is a common figure) and prevented from boiling by a static pressure superimposed on the whole system.

The earliest of such systems was the *Perkins System,* which consisted of a continuous pipe coil about $\frac{7}{8}''$ diameter, part of which passed through a brick-lined furnace as shown in Fig. 200. The head was maintained by means of a sealed expansion vessel partly filled with air, which was compressed as the water expanded.

There are many disadvantages inherent in this system, and it is now entirely obsolete. Much more recently Continental engineers developed the system now known as *High-Pressure Hot Water,* and it has been extensively adopted in this country, particularly for large industrial installations. For this application the principal advantages are:

Mains may be run almost regardless of level, and for long distances if well lagged.

The amount of heat distributed can be adjusted within fine limits.

Circuits for process-heating (such as enamelling ovens) can be maintained at a high and constant temperature, while the space-heating circuits served from the same boilers can easily be modulated in accordance with the outside conditions.

There are no steam-traps or similar accessories to maintain.

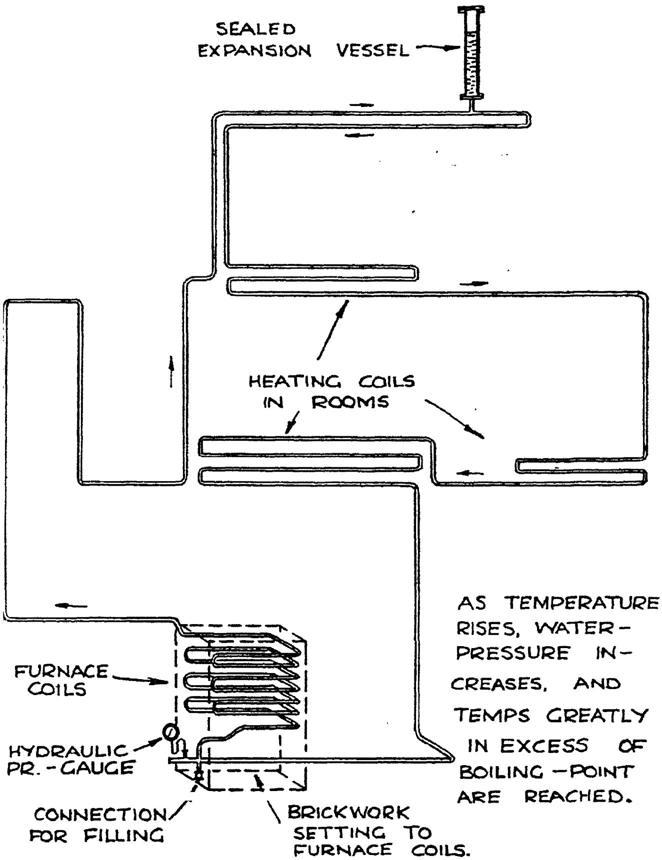


FIG. 200.—Diagram of Perkins System.

Corrosion troubles in the piping are much reduced if not entirely eliminated.

A positive circulation is maintained throughout the system by means of circulating pumps in the boilerhouse.

Pressure on the System—The artificial head on the system may be applied by means of air compressed in a closed head-tank, or by a pump in the feed line. By far the most common method, however, is to use the pressure of the steam generated when the water is heated. This steam accumulates above the water in the boiler, or in a steam-drum above the boiler, the water and the steam thus being in equilibrium at the pressure generated. Reference to the Steam Table on page 351 will show the maximum temperature to which the heating water can be raised for a given working pressure.

To sum up: High-pressure Hot Water as now to be discussed means a

system whereby water is maintained at a temperature in excess of its normal boiling point by the pressure of steam in contact with it, and circulated in a closed circuit by means of centrifugal pumps. For convenience of reference, the letters 'H.P.H.W.' will be used to denote this system.

Saving of Trap Losses—As mentioned above, one of the great advantages of high-pressure hot water is the avoidance of losses associated with steam-traps. It will be remembered from the chapter on Steam that when a steam-trap discharges, the condensate is at the same temperature as the steam and the heat it contains is lost if it runs to waste. If it is piped back to the boilerhouse it must, for use as boiler feed, be below boiling point at atmospheric pressure (212°). The difference of total heat between the two temperatures represents the loss.

Thus, supposing steam is condensed at 100 lbs. per sq. in. gauge, the heat of the liquid (see Table LX) will be:

	309·1	B.T.U./lb.
Return at hot well (at say 200°) - -	168·1	„
Difference - - - - - -	141·0	„
The initial total heat of the steam at 100 lbs. - - - - - -	=1190·7	„

The loss is thus $\frac{141 \cdot 0}{1190 \cdot 7} \times 100$ - - $= 11 \cdot 8\%$

It is possible that some of this loss may be recovered if the condense piping runs inside the building to be heated.

In addition to trap losses, there are trap hold-ups; apparatus going cold because a trap is choked; or lines of steam piping water-logged or air-bound due to trap trouble. These sometimes cause loss of production and waste, as the Engineer may decide to let the open end discharge to waste until he can repair the trap.

Pipe Sizes with H.P.H.W.—It is interesting to compare the sizes of mains required for steam and H.P.H.W. systems supplying the same amount of heat.

Figure 201 shows the relative heat-carrying capacity of various sizes of pipe, and it will be seen that for all sizes the H.P.H.W. system working at 100° F. temperature drop through the apparatus will deliver approximately as much heat as a steam line at 150 lbs. per sq. in.

In drawing curves the following assumptions have been made:

H.P. and L.P.H.W., piping resistance 0·10 inch per foot run.

Steam piping resistance 1 lb. per 100 foot run.

Steam used in all cases at 50 lbs. per sq. inch and the latent heat (912 B.T.U.'s/lb.) at this pressure recovered in the apparatus.

The approximate condense return sizes are given in Curve One, and the L.P.H.W. sizes at 30° F. drop in Curve Five, for comparison. Although the curves appear close together, it will be seen from a consideration of

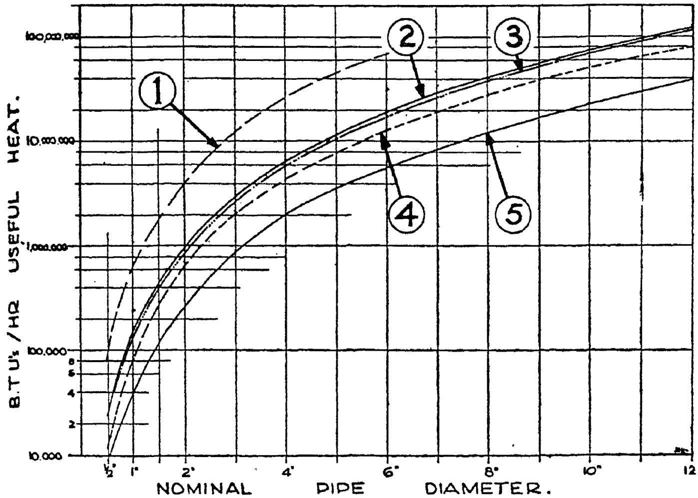


FIG. 201.—Useful heat carried by steam and hot water for pipes of various sizes.

Curve 1. Condense returns.
2. Steam at 150 lbs./sq. inch, and H.P.H.W. at 100° F. temperature drop.
3. Steam at 100 lbs./sq. inch.
4. Steam at 50 lbs./sq. inch.
5. Hot water at 30° F. temperature drop.

(Assumed pressure drops:
water—0·10 inch/foot run,
steam—1 lb./100 foot run.)

the vertical scale that the differences are considerable. Thus, for a 4″ pipe, the heat-carrying capacities, and corresponding steam or water velocities for the assumed pressure drops, are:

		Steam or Water Velocity	
L.P.H.W. at 30° F. drop	2,000,000 B.T.U.'s/hr.	3·5 ft./sec.	
Steam at 50 lbs. -	4,380,000 ,,	115·0 ,,	
,, 100 lbs. -	5,740,000 ,,	78·0 ,,	
H.P.H.W. at 100° F. drop	6,600,000 ,,	3·4 ,,	(at 300°)
Steam at 150 lbs. -	6,750,000 ,,	65·0 ,,	

Conversely, to carry 12 million B.T.U.'s would require 5″ flow × 5″ return H.P.H.W. or 6″ steam × 3″ condense, with steam at 50 lbs. The cost of the former pair of pipes, lagged, is almost exactly the same as that of the latter pair, but the latter also require drainage points, etc., while the H.P.H.W. does not.

Fuel Economy with H.P.H.W.—The heat supply to any item of equipment can be controlled from 100% down to nothing, by the turning of a valve in the H.P.H.W. connection, and the gradation can be uniform over the whole range. This is not possible with steam, except in the case of vacuum systems. Either the steam is full on or right off, since the difference

of heating effect produced by throttling a steam valve is too slight to be noticeable.

This ease of adjustment is particularly important in connection with space heating using pipes or convectors, and with process work such as manufacture of plastics, heating of vats, etc. In fact some processes may be said to depend on the use of H.P.H.W. as a heating medium.

The other aspect of this ease of control is, however, that of fuel economy. The better control coupled with the avoidance of trap losses does in practice give lower consumption. Where conversions have been made from steam to H.P.H.W., savings in fuel of as much as 20% have been reported.

In defence of steam, it must not be assumed that where an old system is brought up to date under H.P.H.W., it could not have been improved a good deal while still retaining steam, but it is clear that there are some ineradicable losses with the latter which hot water avoids.

TYPES OF BOILERS

For smaller systems, i.e. say up to 4,000,000 B.T.U.'s/hr. and using flow temperatures not exceeding about 250° F., cast iron sectional boilers with steam-drums may be used, but for most applications it is necessary to use a steel boiler of one of the following types:

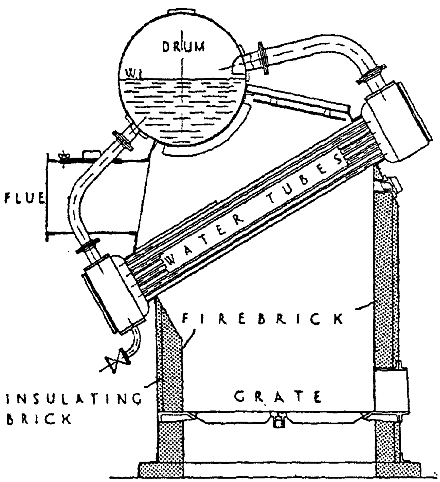


FIG. 202.—'Fraser' Water-tube Boiler.

Vertical Cylindrical such as the Cochran already described (see p. 101), with additional connections for dip-pipes as discussed later.

Water-Tube Boilers primarily designed for steam-supply, and adapted to H.P.H.W. Thus, water-tube boilers, such as Babcock & Wilcox, Stirling, Thompson, Clarke Chapman and others, may be used (generally for large installations) in much the same manner as the Lamont Boiler described later. The water-tube boiler of the 'Fraser' type is also used for H.P.H.W. A section of this boiler is given in Fig. 202.

Horizontal Shell-type—Economic, Super-Economic or Lancashire steam boilers can also be adapted to H.P.H.W., and many are made specially for the latter medium, the shape and arrangement of the boiler being the same as for steam.

Fig. 203 shows a typical arrangement of a boilerhouse, using six *Super-*

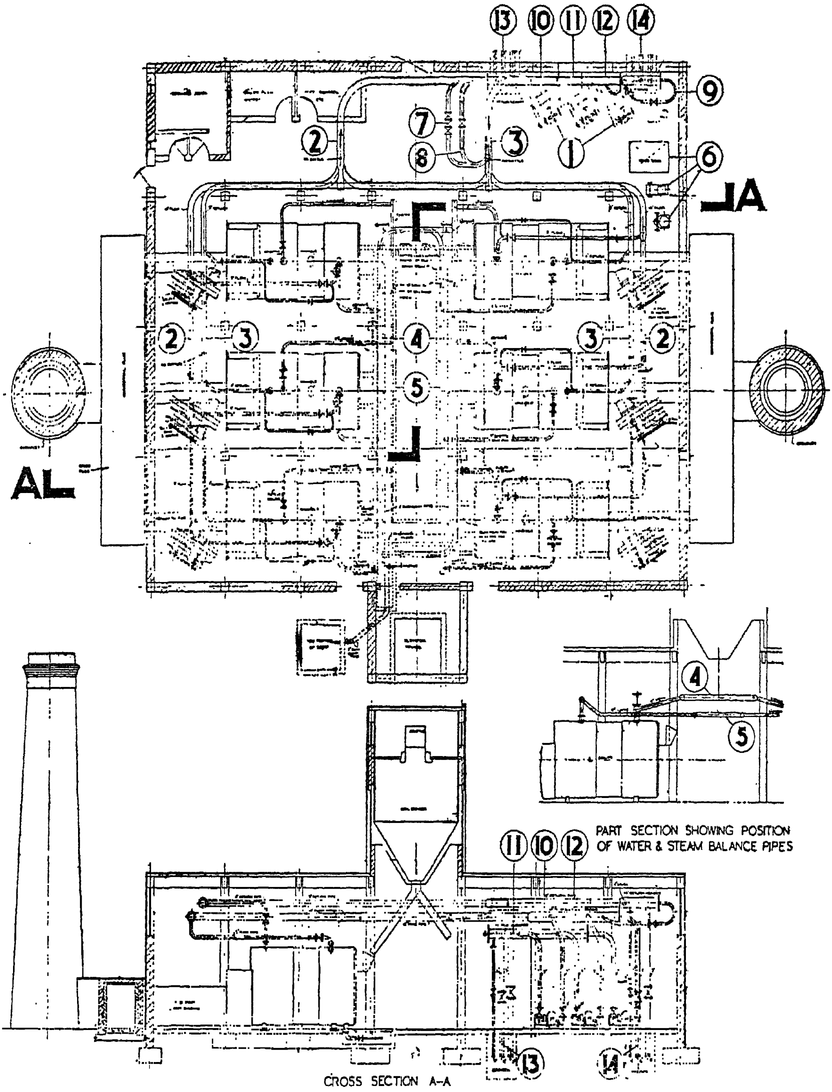


FIG. 203.—General arrangement of boilerhouse with six (Danks of Netherton) Super-Economic H.P.H.W. Boilers. 1. Circulating pumps. 2. Flow main. 3. Return main. 4. Steam balance main. 5. Water balance main. 6. Feed pumps and tank. 7. Thermostatic mixing valve. 8. Hand-controlled mixing connection. 9. Secondary mixing connection. 10. Pump suction header. 11. Mixed-water flow header. 12. Return header. 13. Outgoing flow mains. 14. Incoming return mains.

Economic Boilers, each rated at 12 million B.T.U.'s/hr. at 80 lbs. per sq. inch maximum working pressure. It will be seen that the boilers are provided with induced draught fans and are connected to common horizontal brick flues and chimneys. The automatic stokers, of which there are two per boiler, are not shown in the drawing. Various other features, such as the flow and return and balance-piping, also the circulating pumps, will be seen in the drawing, and will be referred to later.

The mountings for a boiler of this type, used for H.P.H.W., consist of:

Safety Valves, as for steam boiler of corresponding duty.

Duplicate Gauge Glasses. These are somewhat longer than would be used

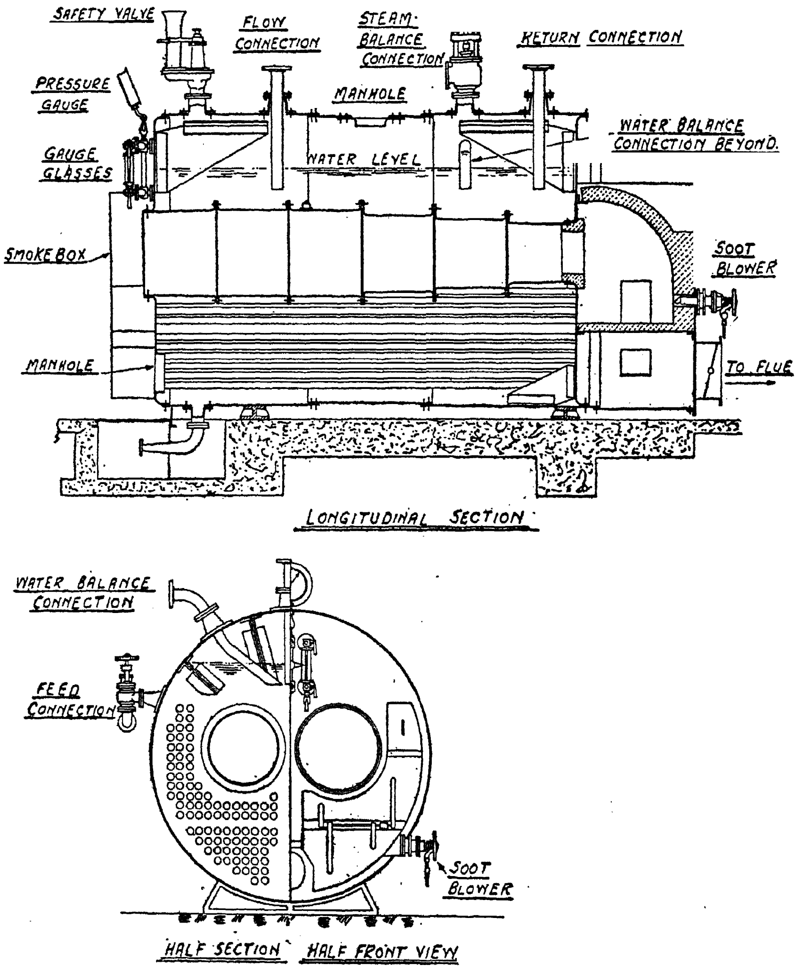


Fig. 204.—Danks Super-Economic Boiler (Patent No. 429383) arranged for high-pressure hot water system.

on a steam boiler, and are so placed as to cover the whole range of water level in the boiler.

Pressure Gauge, showing the steam pressure in the upper part of the boiler.

High and Low Water Level Alarm of the standard type as used on steam boilers.

Connections are provided on the boiler for flow and return connections with stop valves to each, also for steam and water balance-pipes. Fig. 204 shows the details of one of these boilers, and the following points should be noted.

The steam balance pipe is taken off the top of the boiler in the ordinary way, as also is a connection (not shown in the drawing), for the steam supply to the soot blowers and to the steam-driven feed pump. The water balance connection is taken off the top of the boiler with a dip-pipe.

The flow and return connections both come near the top of the boiler and are carried down below the water line by means of *dip-pipes.* These may either be open-ended pipes, or may have special ends to improve the distribution of the water within the boiler. It is found, for instance, that if the flow pipe has an open end situated directly over the boiler flues, a good deal of steam is picked up off the latter and carried into the pipe-work. To obviate this a shoe consisting of a half-round piece of pipe is fitted, so that the water has to pass into the top half of the shoe, while the bottom half deflects the rising bubbles of steam, which then pass to the steam space.

Lamont Type—This type of high velocity water-tube boiler is designed specially for H.P.H.W., and is illustrated in Fig. 205. The boiler consists of a nest of $1\frac{1}{4}''$ bore, weldless steel tubes, enclosed in the upper part of a firebrick-lined and insulated rectangular casing, the lower part of which forms the grate and combustion chamber. The smoke-outlet is at the top, and the boiler can be worked on natural or on induced or forced draught, the first type being shown in the figure.

Firing may be by hand, or by automatic stoker as illustrated. The shape of the combustion chamber can be adapted to any method of firing, such as underfeed, sprinkler, chain-grate or coking stokers, and to fuel oil, gas, or waste heat.

The lower end of each tube is connected to a return header, seen on the right near the floor. If there is only one boiler, then the upper end of each tube is taken into a cylindrical drum generally mounted on top of the boiler. The latter is thus always full of water, and the water-surface in contact with the steam is near the centre of the drum.

If there is more than one boiler in the battery, the upper ends of the tubes deliver to a flow header exactly like the return, and a single connection is taken from each flow header to a common drum, which is generally situated at a level above the tops of the boilers, for convenience of venting. The drum may, however, be at a lower level if necessary to fit into an existing boilerhouse, or for other reasons.

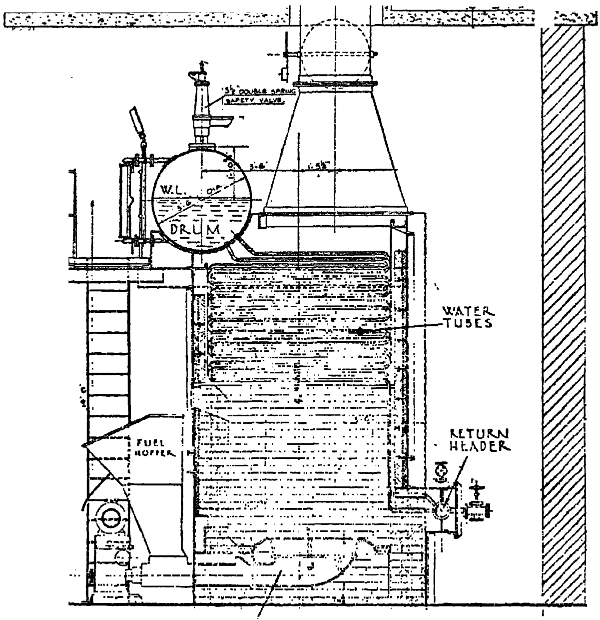


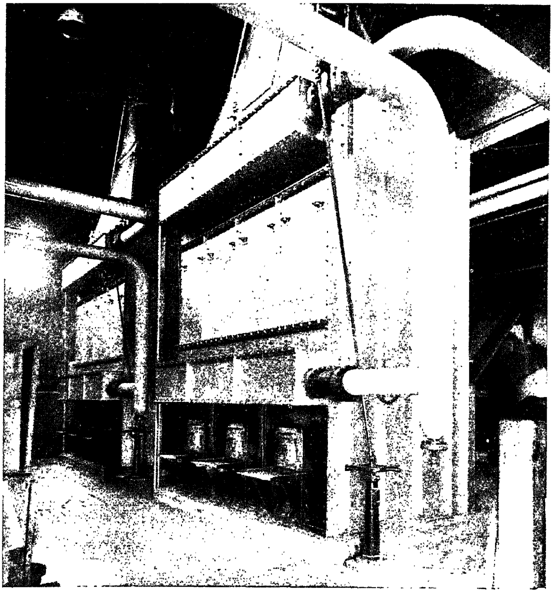
FIG. 205.—Section of a Lamont Boiler.

The arrangement of a number of boilers connected to a common drum is shown diagrammatically in Fig. 206, and the application of this layout to an actual installation will be seen in Fig. 207, which shows a battery of three boilers connected to the left-hand drum, and two similar boilers connected to the right-hand drum with provision for a third on this side, so that the ultimate scheme will consist of a double boilerhouse, of which either half can be worked separately, or the two halves run together as one unit. The boilers in question are rated at 12,000,000 B.T.U.'s per hour each, at 80 lbs./sq. inch maximum working pressure.

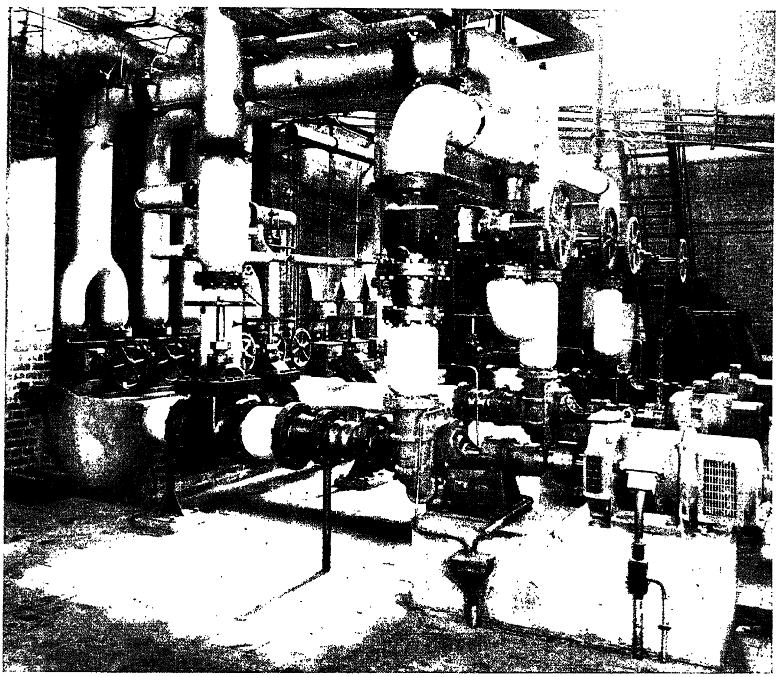
Fig. 207 shows only the piping between boilers and drums. For arrangement of the circulating piping see Fig. 206.

Mountings—The usual fittings and mountings for a range of Lamont boilers are:

(*a*) *On each boiler*, a spring-loaded relief valve of nominal size ($1\frac{1}{2}''$-$2''$)



(a)



(b)

Plate XXI (*a*). *Lamont H.P.H.W. boilers on large industrial heating and process installation.* (*b*) *Circulating pumps and interconnecting pipe work* (*Sulzer Ltd.*) (*see pp. 363 and 372*)

Photos: courtesy of *Industrial Heating Engineer*

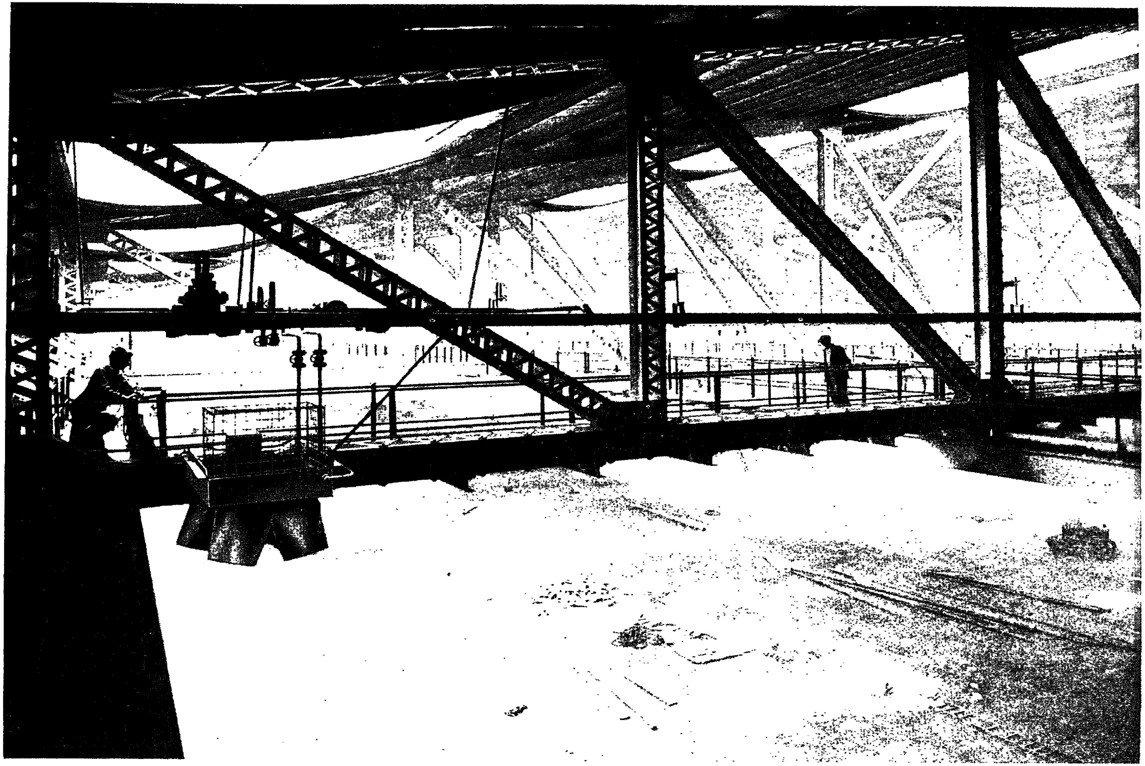


Plate XXII. View showing 240 ft. span aircraft construction shed heated by vertical discharge H.P.H.W. Unit Heaters (see p. 380)

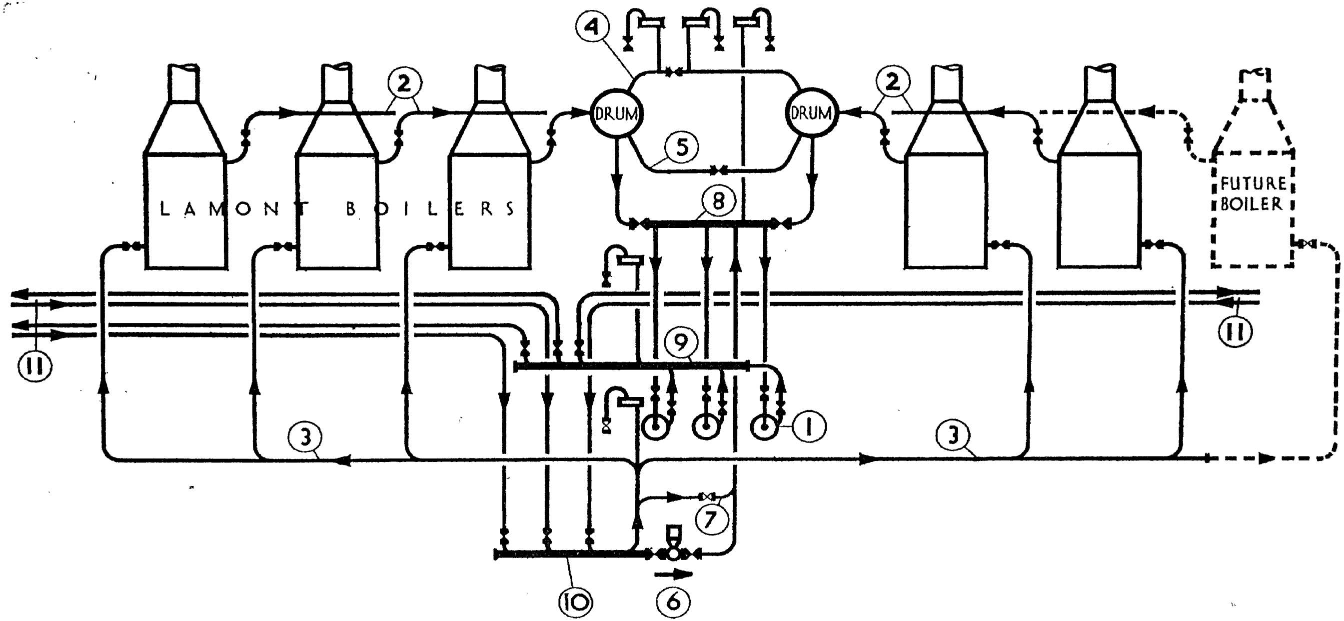


FIG. 206.—Diagram of main circulating piping for H.P.H.W. system served from the boiler plant shown in Fig. 207.

1. Circulating pumps. 2. Boiler flow mains (each run separately to drum). 3. Boiler return mains. 4. Steam balance-pipe. 5. Water balance-pipe. 6. Thermostatic mixing-valve. 7. Hand-operated bye-pass. 8. Mixed-water header (pump suction). 9. Flow header (pump discharge). 10. Return header. 11. Heating circuits to buildings. Air bottles where shown (cf. Fig. 214).

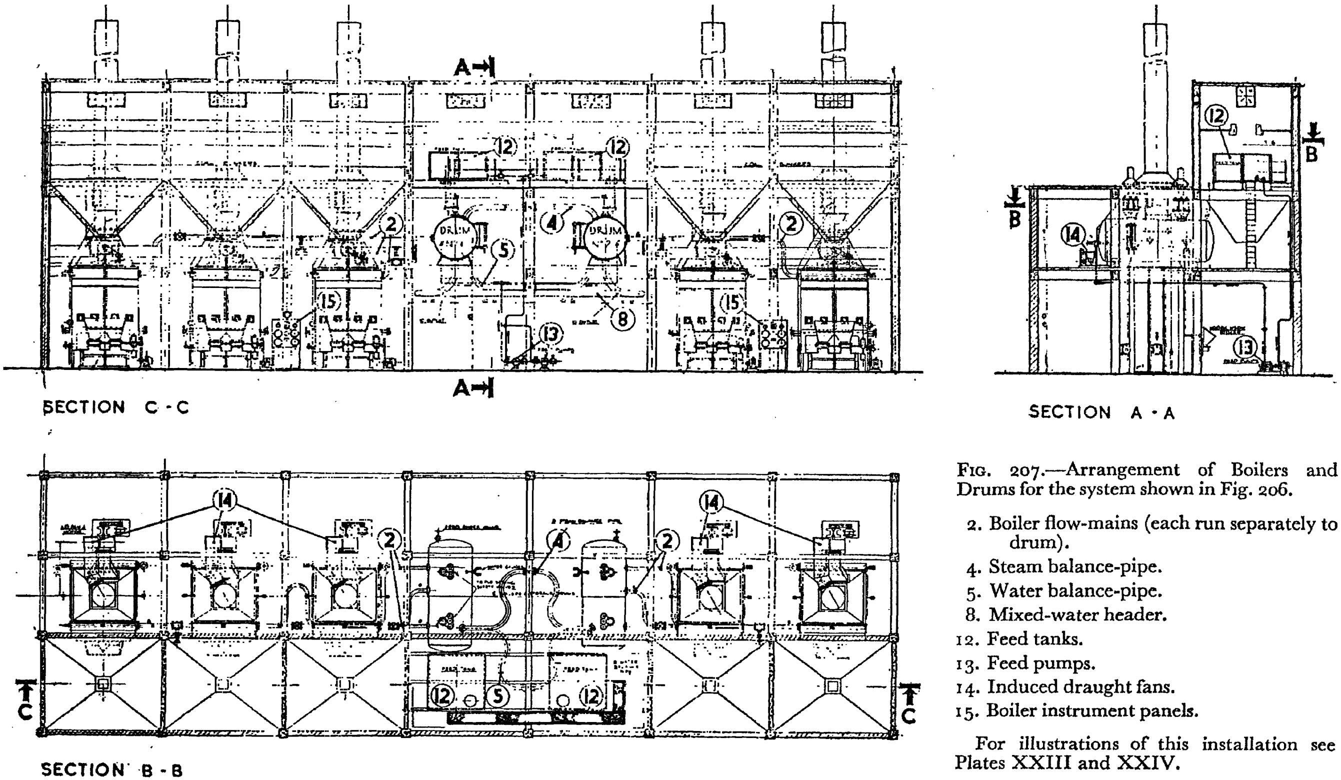


FIG. 207.—Arrangement of Boilers and Drums for the system shown in Fig. 206.

2. Boiler flow-mains (each run separately to drum).
4. Steam balance-pipe.
5. Water balance-pipe.
8. Mixed-water header.
12. Feed tanks.
13. Feed pumps.
14. Induced draught fans.
15. Boiler instrument panels.

For illustrations of this installation see Plates XXIII and XXIV.

to relieve the hydraulic pressure in the event of the water being heated with the valves closed.

Main parallel-slide stop-valves on flow and return.

Draincock on return header.

Aircock on flow header.

There is also provided in some instances a Differential Pressure Gauge to show the pressure loss across the boiler.

(*b*) *On the drum*, duplicate gauge-glasses with cocks.

Combined feed-check valve.

Airbottle and blow-down cock.

High- and low-water alarm, of the float type, arranged to ring a bell, or electrically connected to cut out the automatic stokers on the boilers.

Steam safety-valves and vent-pipes. These valves are to Factories Act requirements in accordance with the steam-raising capacity of the boilers connected to the drum.

Steam stop-valves as required, for steam supply to soot-blowers, etc.

Plate XXI (*a*) (facing p. 360) shows the backs of two large Lamont boilers, and the connections thereto. The flow and return headers will be seen, both on the back of the boiler in this case, with remote operation of the flow valve. The three doors seen at the bottom of the boiler are for removal of ash and clinker.

BALANCING OF BOILERS

When a number of boilers are connected together for H.P.H.W. heating, it is essential that the water levels in each boiler shall be the same, and to this end steam and water *balance-pipes* are provided, connecting up the boilers on to a common main. Normally, no water or steam will flow in these pipes unless required to do so to maintain the common water level. These pipes will be clearly seen in Fig. 203, in which each boiler has a 6 in. steam pipe connected on to an 8 in. loop-main in the centre of the boilerhouse, and a 5 in. water balance-pipe connected to a 7 in. loop.

If two or more boilerhouses at a distance were used to feed into a common H.P.H.W. system, the sites would have to be chosen so that the boilers could be at identical water level.

In the case of the Lamont boilers in Fig. 207, the water surfaces occur in the drums, as already described, and the balance-pipes—8 in. steam and 8 in. water—are provided between the top and bottom respectively of the two drums. If all five boilers had been connected to one drum, as might have been done, no balance-piping would have been required.

It is also desirable that the steam spaces of the boilers or drums shall be of similar volume in each case, and the boiler bases should be constructed at a common level, and the boilers set identically upon them.

Balance-Pipe Sizes—The sizing of balance-pipes is empirical, and is not critical provided the pipes are of ample size to allow surge of water and

steam without appreciable frictional resistance. The following table gives a guide to the usual sizes for a system working at 80 lbs./sq. inch.

Boiler or Drum Rating, in B.T.U.'s per hour	Nominal size of Balance-Pipes
2,500,000	3″
5,000,000	4″
10,000,000	5″
15,000,000	6″
20,000,000	7″
30,000,000	8″

NORMAL WORKING PRESSURES AND TEMPERATURES

The pressure at which the system is worked will be controlled by the use to which the heat is put, and may be considered under the following headings:

For heating by means of unit heaters, the outgoing water must be hot enough to make effective use of the heating surface in the units without giving an excessive and uncontrollable air temperature.

For process work, such as ovens, the temperature is determined by the nature of the process and may be very high, in which case there is no option but to work the boiler at a high enough pressure to give the required temperature.

For steam raising in a water/steam calorifier, the H.P.H.W. must be at a temperature in excess of the saturation temperature of the steam required from the calorifier.

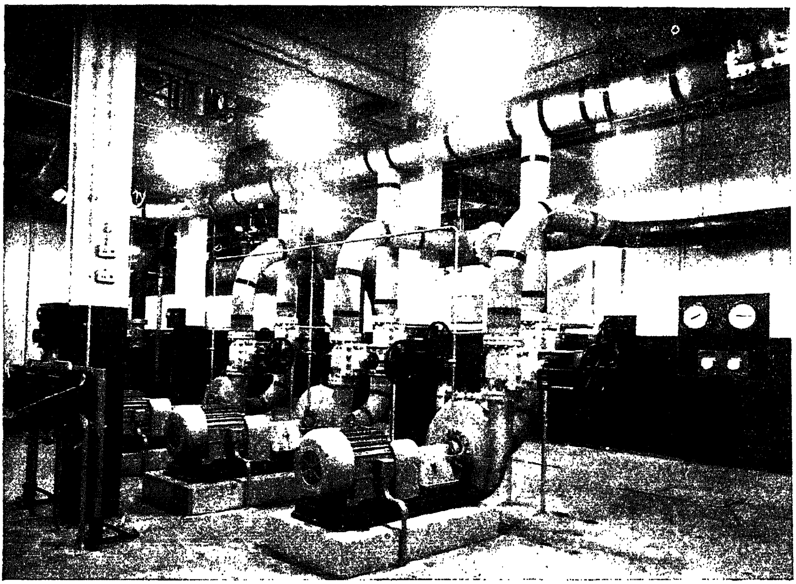
Subject to these considerations the working pressure of the system should be kept as low as possible, and for ordinary heating schemes, without long runs of external mains, 80-120 lbs. per sq. in. gauge is usual. This pressure range corresponds to a temperature range of 324°-350° at the boilers, and a consideration of Table LX will make it clear that relatively little temperature rise is gained by increasing the pressure above the range given; for instance, at 200 lbs. per sq. in. the saturation temperature is 388°, i.e. a rise of only 38° for a pressure increase from 120 to 200 lbs. Further, boilers for pressures like 200 lbs. per sq. in. are more expensive, as they have to be made of thicker plates and with heavier flanges, mountings, valves, etc.

In the case, however, of a system serving a very large site, or group of sites, where the runs of external mains are long, and the heat losses correspondingly high, the extra temperature available at 200 lbs. pressure may be worth while, so as to deliver the water to the apparatus at an economical temperature of use, whilst permitting also a greater temperature drop between flow and return with more heat carried for a given volume of water.

MIXING OF WATER

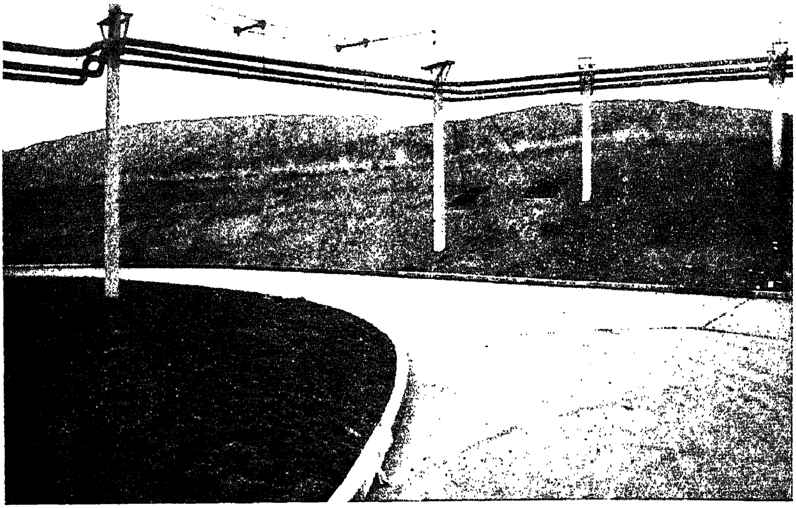
Fig. 208 shows diagrammatically the main piping and circulating pumps for the Super-Economic Boiler system already described, and illustrated in Fig.

Lamont boilers with Hodgkinson stokers



Circulating pumps

Plate XXIII. High Pressure Hot Water system heating a large Government depot, installed to the Authors' designs by Messrs. Richard Crittall, Ltd. (For diagram of the system see fig. 206)



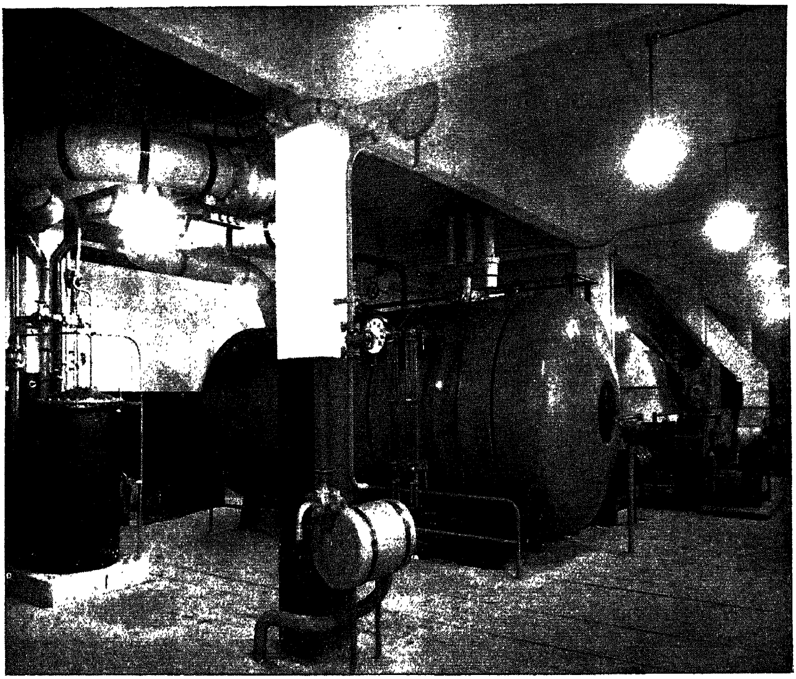


Plate XXIV. Top, High Pressure Hot Water mains showing road crossing and supports. Bottom, Steam drum connected to the boilers shown overleaf. Induced draught fans, air bottle and H.W.S. calorifier will be noted

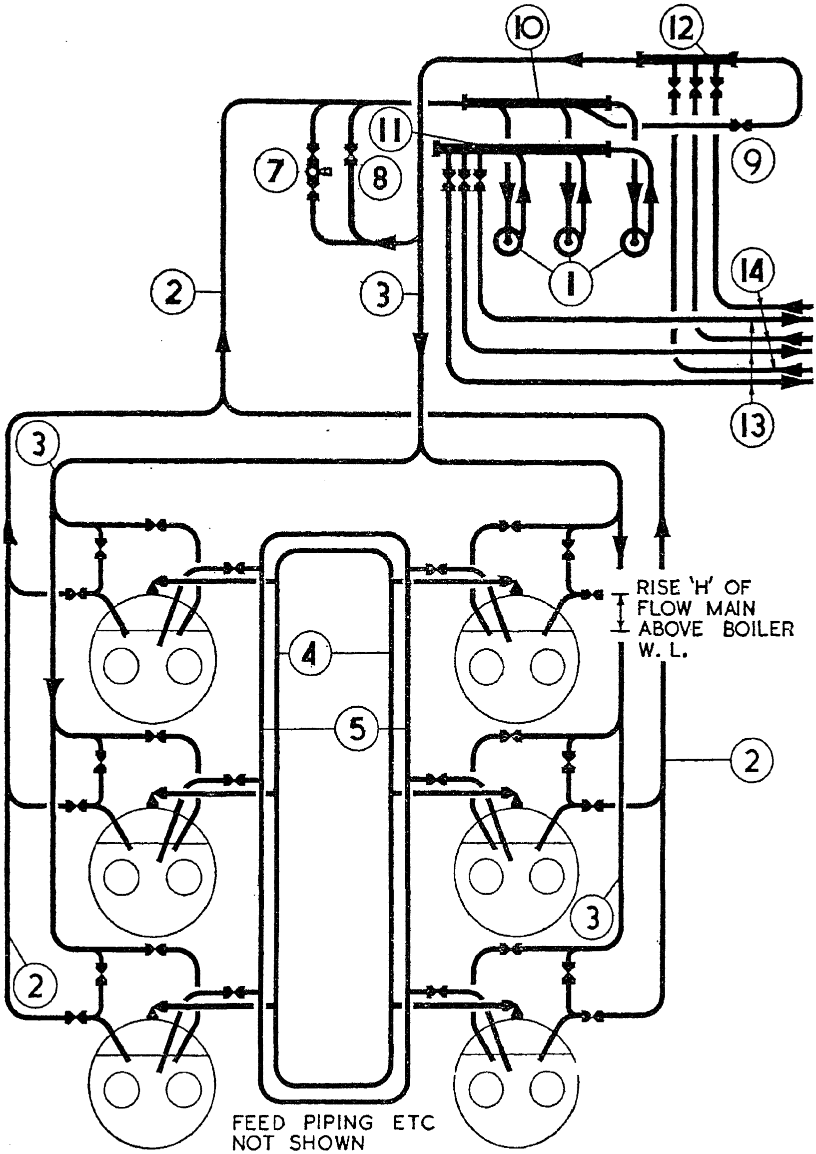


Fig. 208.—Diagram of the main piping and circulating pumps for the Super-Economic Boiler system described and illustrated in Fig. 203. 1. Circulating pumps. 2. Flow mains. 3. Return mains. 4. Steam balance main. 5. Water balance main. 7. Thermostatic mixing valve. 8. Hand-controlled mixing connection. 9. Secondary mixing connection. 10. Pump suction header. 11. Mixed-water flow header. 12. Return header. 13. Outgoing flow mains. 14. Incoming return mains. (The numbering corresponds with that on Fig. 203.)

It will be noted that the pumps are in the flow main, which is the arrangement usually adopted for this type of system, and the one considered in the discussion which follows.

If the working pressure of the boiler is, say, 80 lbs. per sq. in., this is the pressure at the water level, and corresponds to a boiler temperature of 324° F. The flow pipe rises above this level by a height '*h*' ft., which corresponds to a static head of ·*42h* lbs. per sq. in. at 200° F. This reduction of pressure would cause the water, which is still at 324°, to boil and generate steam in the flow pipe, thus stopping the circulation or, at any rate, causing serious hammering in the pipes. To obviate this a bye-pass connection of about 2 in. bore, with a regulating valve, is provided across the boiler, as shown, so as to inject some of the cooler return water into the flow pipe and reduce its temperature to one at which it will not boil under the reduced pressure.

Further along the flow pipe the water enters the circulating pump, which may be generating a head of something like 60-80 ft. Fig. 209 shows

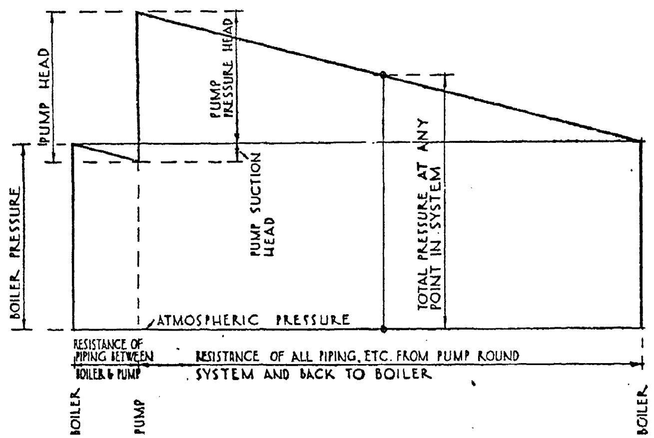


FIG. 209.—Pressure diagram for typical H.P.H.W. system.

how this head is dissipated throughout the system. It will be seen that the head consists of a suction or negative head between the boiler and the pump, and a pressure or positive head from the pump, round the heating system and back to the boiler. The relative magnitude of these two heads depends on the pipe sizing, and the pressure head is, of course, much the larger. The negative head will, however, give further reduction of pressure in the flow pipe with consequent ebullition in the pump suction header unless steps are taken to prevent this.

Mixing Connection—The method usually adopted to prevent ebullition is to provide a mixing connection or bye-pass, as shown in Figs. 206 and 208, so as to inject cool return water into the pump suction header. Thus, in the system shown in Fig. 203, the maximum boiler water temperature

is 324° (80 lbs. per sq. in.), while the maximum flow water temperature leaving the pumps is 300° F., which corresponds to 52 lbs. per sq. in. The difference, i.e. 28 lbs. per sq. in., is available to offset the effects of the reductions in pressure described above.

In the system shown in Fig. 207 the former type of pressure reduction, i.e. that due to the flow main rising off the boiler, does not apply, since the water surface is in the drum, which is at the highest point, and the pump suction pipe drops away from this. The second reduction of pressure, that at the pump suction, is common to any system.

Size of Mixing Connection—The sizing of the mixing connection will depend upon the resistance to flow through the boiler. If the mixing connection is over-sized it will be possible, by fully opening the mixing valve, to bye-pass an undue proportion of the circulated water so that too little is passing through the boilers. This might be dangerous as it would cause rapid generation of steam, which would lower the water line and might uncover the tops of the boiler flues, in the case of the Super-Economic boiler, or cause overheating of the water tubes, in the case of the Lamont boiler. It will be seen, therefore, that the higher the resistance to flow of water through the boiler itself, the smaller should be the mixing connection, so as to provide a comparable resistance.

Thermostatic Control of Mixing—The mixing connection will always have a valve for control purposes, and this valve may be hand-operated, or it may be thermostatically controlled in accordance with the heat demands. In the latter case a convenient control is by means of a Variostat (see p. 132), having one bulb in the flow main and one in the outside air, the temperature of which is a measure of the heat requirements of the building served.

If thermostatic control is used, there is a possibility of the mixing valve being fully closed, and, therefore, a separate mixing pipe in parallel with the main one is generally provided, so that even under the coldest weather conditions, when no mixing is called for by the Variostat, a controlled amount of mixing may be done on the second bye-pass, to avoid ebullition. This second bye-pass will be seen in Figs. 203 and 208, between the return header and the pump suction header. In this system the main thermostatically controlled mixing pipe (7) is 6 ins. bore and the secondary one (9) is 4 ins.

There is also shown in Fig. 208 a hand-controlled bye-pass (8) of the same size as the thermostatic valve. This is provided solely as a standby to the latter, which may be isolated or even removed for repair and cleaning, while retaining full control of the system.

Zoning at Different Temperatures—If the nature of the job requires it, a separate mixing valve may be provided for controlling each of two or more zones, working at different temperatures, into which the system may be divided. A similar application of the same principle is when heating and process mains are served from the same boilers, in which case the process mains, which are worked at a constant flow temperature, have a hand-set

mixing connection which is not altered, while the heating circuits are separately controlled in accordance with the weather.

TABLE LXI

PROPERTIES OF WATER AT HIGH TEMPERATURES

Pressure lbs./sq. in. Gauge	Boiling Point °F.	Specific Heat B.T.U.'s/°F.	Density lbs./cu. ft.	Specific Volume cu. ft./lb.*	lbs./gallon
Atmospheric	212°	1·006	59·82	·01671	9·59
2·5	220°	1·007	59·62	·01677	9·56
5	227°	1·008	59·45	·01682	9·54
6	230°	1·009	59·37	·01684	9·52
10	239°	1·012	59·10	·01692	9·48
12·6	245°	1·014	58·91	·01697	9·45
15	250°	1·015	58·79	·01700	9·42
17·8	255°	1·017	58·69	·01704	9·41
20	259°	1·018	58·56	·01708	9·40
23·8	265°	1·019	58·42	·01712	9·37
25	267°	1·020	58·35	·01714	9·36
27·1	270°	1·021	58·22	·01717	9·33
30	274°	1·022	58·11	·01721	9·32
34·5	280°	1·023	57·94	·01726	9·29
35	281°	1·023	57·91	·01727	9·29
38·5	285°	1·024	57·77	·01731	9·27
40	287°	1·025	57·69	·01733	9·25
42·8	290°	1·026	57·59	·01736	9·24
45	292°	1·026	57·54	·01738	9·23
47·4	295°	1·027	57·44	·01741	9·21
50	298°	1·028	57·32	·01744	9·18
52·3	300°	1·029	57·2	·0175	9·2
55	303°	1·030	57·1	·0175	9·2
60	308°	1·031	56·9	·0176	9·1
65	312°	1·032	56·8	·0176	9·1
70	316°	1·033	56·7	·0176	9·1
75	320°	1·035	56·5	·0177	9·0
80	324°	1·036	56·4	·0177	9·0
90	331°	1·038	56·3	·0178	9·0
100	338°	1·041	56·1	·0178	9·0
110	344°	1·043	55·8	·0179	8·9
120	350°	1·045	55·6	·0180	8·9
140	361°	1·050	55·2	·0181	8·8
160	371°	1·054	54·9	·0182	8·8
180	380°	1·060	54·4	·0184	8·7
200	388°	1·066	54·1	·0185	8·7
FEED WATER	50°	1·001	62·41	·01603	10·00

* This column taken from Callender's *Steam Tables*, 1939, published by Edward Arnold.

EXPANSION OF THE WATER

When the system is heated up, the whole of the water contained in it will expand, causing a rise of water-level in the boiler or drum, and this

rise may be allowed to take place within certain limits, but the conditions should be carefully checked to see what are the safe limits.

Table LXI gives the density and specific volume of water at various pressures and temperatures, from which the magnitude of the expansion may be calculated.

Suppose that the system served from the boilers in Fig. 207, together with the boilers themselves, contains 300,000 lbs. of water. The volume of this, when cold, will be:

300,000 × ·01603 - - - - - -	=4809 cub. ft.
Volume at mean temperature of 230° -	=5052 ,,
,, ,, ,, 300° -	=5250 ,,
Increase from cold to hottest working condition	= 441 ,,
Increase from medium condition to hottest working condition - - - -	= 198 ,,

It is uneconomical to cater, in the capacity of the drums, for the former variation, which occurs very infrequently and can be met by feeding water into the system or by bleeding it off; but the latter variation can, and should, be provided for without the necessity for changing the water in the system.

If a variation of water-level in the drums of one foot is allowable, the water-line area must be 198 sq. ft., and two drums each 6′ 0″ diameter × 16′ 6″ long would have this area with the water-line near the centre.

The contents of the system can be estimated by taking off from the drawings the total length of piping of each size, and using Table XL (p. 216) for the volume of unit length of piping. The volume of the boilers, drums (half-filled), unit heater batteries, etc., must be added in.

Where the water is drawn from the pressure vessel by means of dip-pipes, the permissible variation in water-level should not be too large, as otherwise, when the level is highest, the flow-pipe will be drawing from a level where the water is not as hot as it is nearer the surface, with consequent unsatisfactory operation.

CIRCULATING PUMPS

The circulating pumps are one of the most important items in any H.P.H.W. system, and the type and capacity should be very carefully considered.

Fig. 210 shows one of the types of pump available for this duty. It is of the end-suction centrifugal type direct-coupled to an electric motor, which may be of constant or variable speed pattern, generally the former.

Points usually taken care of in the design of circulating pumps are:

Suction connection designed to avoid sudden changes of velocity, and consequent cavitation.

Casing generally of cast steel to withstand high pressures.
Gland cooled by water fed under pressure and with visible discharge into a tundish.
Bearings suitable for high temperature.

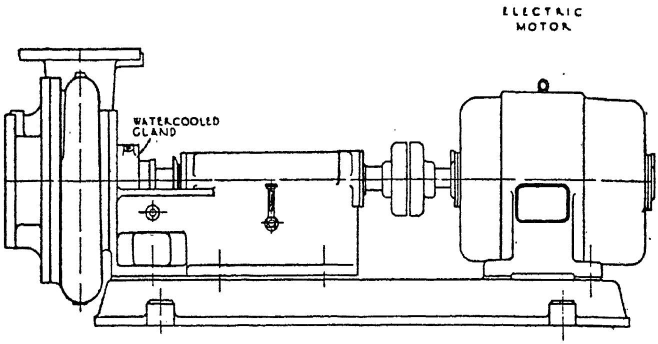


FIG. 210.—H.P.H.W. Circulating Pump (Hobdell).

Pump Duty—The pump-head should be calculated for the resistance of the whole of the piping under conditions of maximum load, including boiler-house piping, plus the loss through the heating apparatus and the boilers, and including a margin to cover the losses at the entry and discharge from the pump. Heads of up to 100 ft. are common in large installations.

The capacity of the pump is generally expressed in gallons per minute. It will be seen from Table LXI that at the usual working temperatures a gallon of water does not weigh 10 lbs., and the conversion-factor must be calculated for each case. Thus, for a system having a designed temperature range of 300°-200° F., and a load of 30,000,000 B.T.U./hr., the pump capacity would be calculated as follows:

Total heat per lb. of water at 300° = 269·7 B.T.U.'s.

" " " " $200° = \dfrac{168{\cdot}1}{101{\cdot}6}$ "

Pump duty $= \dfrac{30{,}000{,}000}{101{\cdot}6} = 295{,}000$ lbs./hr.

Water at 300° (the temperature at which it is handled by a pump in the flow)

Spec. vol. = 0·0175 cub. ft./lb.

$$\frac{{\cdot}0175 \times 6{\cdot}24}{60}$$

TABLE LXII

B.T.U.'S PER HOUR CARRIED BY ONE GALLON PER MINUTE PASSED BY PUMP IN FLOW

Return Temperature °F.	Flow Temperature in degrees F.							
	230	240	250	260	280	300	325	350
200	17,300	22,850	28,600	34,200	45,200	55,800	69,200	82,100
210	11,500	17,200	22,850	28,500	39,600	50,400	63,800	76,600
220	5,750	11,500	17,150	22,800	33,900	44,800	58,200	71,200
230	—	5,750	11,470	17,150	28,300	39,300	52,800	65,900
240	—	—	5,730	11,420	22,650	33,700	47,300	60,500
250	—	—	—	5,710	17,020	28,150	41,800	55,100
260	—	—	—	—	11,400	22,550	36,300	49,600
270	—	—	—	—	5,700	16,900	30,750	44,200
280	—	—	—	—	—	11,280	25,200	38,700
290	—	—	—	—	—	5,640	19,600	33,200
300	—	—	—	—	—	—	14,050	27,800

For convenience of calculation, Table LXII shows the amount of heat in B.T.U.'s per hour corresponding to one gallon of water per minute passing through the pump, for various flow and return temperatures. The pump is assumed to be handling the water at the flow temperature.

The pump duty remains approximately constant under conditions of mixing, since the bye-pass pipe is taken into the pump suction, so that all the water passed to the system is handled by the pump, whether it passes through the boiler or through the bye-pass.

Standby Pumps—If the pump stopped, no heat could be removed from the boilers, and action would have to be taken at once to prevent the boilers steaming, especially if they were at the time under heavy load. To obviate as far as possible the risk of pump failure, it is usual to provide at least two pumps connected in parallel and provided with isolating valves, so that either or both may be used. This arrangement has the additional advantage of allowing pump and motor maintenance-repairs without stopping the system.

If two pumps are provided, each may be sized for the full load of the system, with the other purely as a standby. A more economical method is to size each pump for two-thirds or three-quarters of the maximum duty, so that in normal weather one pump would suffice, with the possibility of using both pumps in severe weather. Similarly, if three pumps are provided, as in the systems shown in Figs. 203 and 207, it is sufficient if two of them are able to carry the designed load.

When only one of the pumps is used, the resistance head will fall, due to the decrease in the amount of water circulated. This drop in head will allow the pump to deliver more water, until it reaches a state of equilibrium at some new point on its characteristic curve. Fig. 211 shows a

typical characteristic for a H.P.H.W. pump. The motor should be sized so that it is not overloaded at any operating point on the curve.

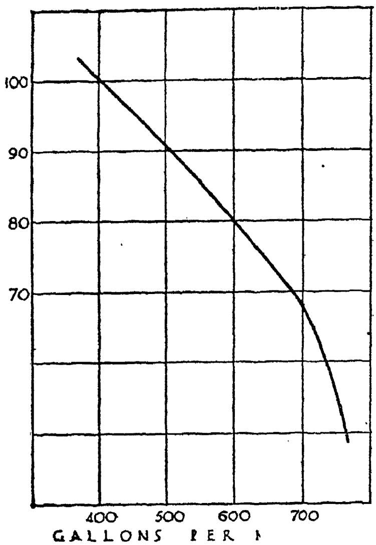


FIG. 211.—Typical Circulating Pump Characteristic.

Plate XXI (*b*) (facing p. 360) and Plate XXIII (facing p. 364) show two typical arrangements of circulating pumps and piping. The flow and return thermometers, and the pump pressure gauges, will be noted in the latter plate.

COLD WATER FEED TO THE SYSTEM

Feed Pumps—The expansion of the water on heating up has been discussed. Conversely, when the system cools down the volume of the water will decrease, and extra water may have to be supplied so as to maintain the water-level. Boiler feed-pumps are, therefore, a necessary part of any H.P.H.W. installation, and may either be of the reciprocating steam-driven simplex or duplex type as used for steam boilers, or they may be motor-driven centrifugal type, or even in certain installations, motor-driven reciprocating type. Centrifugal pumps will be seen in Fig. 207 drawing water from overhead tanks at high level and delivering it to the drums. The reason why motor-driven pumps are often specified is that steam for driving the pumps must come from the heating system and may not be available when it is required, if the boilers are only just being started up. An arrangement often adopted is to have one steam-driven pump and one electrically-driven.

The connection to the boiler or drum is of the ordinary type, using a combined feed- and check-valve. Feed water injectors are not recommended as a substitute for feed pumps on H.P.H.W. systems.

In addition to making-up to compensate for contraction, on the occasions when this is necessary, the feed pumps make good any actual loss of water from the system due to one or more of the following causes:

- Leakage at pumps and valve-spindles.
- Blowing-down of boiler or drum.
- Partial emptying down for repairs.
- Venting of mains and apparatus, which always removes some water and steam as well.
- Use of steam for soot-blowers, for driving the feed pumps, or for steam-jets on certain types of mechanical stoker.

Of these the last named are by far the most important. In normal running of a system in which no steam is removed for any purpose, the daily make-up is very small, and the feed-pumps would be sized from consideration of the initial filling-up and end-of-season make-up. If steam is taken from the system the amount used must be ascertained and the feed-pumps sized accordingly. In any case, the size will be much smaller than for the corresponding steam system, in which the feed-pump has to handle the whole of the returned condensate, in addition to making-up losses at traps, etc.

Cold Water Storage—The feed-pumps draw from storage tanks, which may be situated at any convenient level provided the pumps are primed. The tanks should be of ample capacity, to guard against any possibility of the boilers being starved of water in an emergency.

Treatment of Feed Water—The question of softening or other treatment of the feed water to obviate scaling of the pipes and boilers, is one on which no definite rule can be laid down. Obviously its importance depends upon the quantity of feed water used, and its chemical analysis, but it may be said in general that treatment is not likely to be necessary for totally closed systems, i.e. those having no draw-off of steam, if the water is of normally good quality as regards hardness and acidity.

For systems where steam is used, however, the problem is very different. In a large system having both soot-blowers and steam jets for automatic stokers, the daily water consumption under full load conditions might reach 1500 gallons, and assuming a heating season of 200 days at an average operating factor of 0·5, 150,000 gallons of water would be used, which would be equivalent to several complete changes of water in the entire system. Without water treatment, all the salts contained in this water would be left in the system, either in solution or as a deposit, since the steam cannot carry any salts away with it. Feed water containing salts must, therefore, be treated, i.e. the salts removed, before use. The *lime-soda* process is often used where the problem is only one of dealing with carbonates. It must be borne in mind, however, that although this process reduces the salts, it does not entirely remove them, and the increase in the salt-content of the system is not therefore eliminated but merely reduced.

Undoubtedly the ideal way from the point of view of eliminating excess solids from the system is to feed into it exactly what is taken out; that is, distilled water; and a softening process using *carbonaceous zeolite* as the reagent in a base-exchange system, is available. This does produce what is practically solid-free water, but unfortunately the plant is more expensive than the lime-soda, and has to be regenerated with sulphuric acid and caustic soda, both requiring special vessels and piping, and both being substances requiring care in handling.

Further, ordinary distilled water is corrosive to steel pipes unless freed of the dissolved oxygen and carbon dioxide, and this adds further

complication to the plant, and requires skilled and accurate control for its proper operation.

Another approach to the problem of removing steam from the system is to provide a separate small steam boiler for all steam services, using treated water if required, combined with a system of continuous blow-down. This would localize the scaling troubles in the steam boiler instead of spreading them all round the main system.

BOILERHOUSE INSTRUMENTS

The number and type of instruments to be provided in the boilerhouse depend on the size of the installation, the degree of control required, and of course on what can be afforded, since expenditure on instruments can reach a high figure. The following instruments, however, are generally considered essential:

(*a*) Boiler or drum pressure-gauge.
(*b*) Boiler thermometer.
(*c*) Flow-main thermometer.
(*d*) Return-main thermometer.
(*e*) Pressure-gauges on pump suction and delivery.

Other instruments usual in larger systems, roughly in order of usefulness, are:

(*f*) CO_2 indicator.
(*g*) Boiler or drum pressure recorder.
(*h*) Flue-gas temperature indicator.
(*i*) Draught gauge.
(*j*) Flow and return thermometers on each outgoing circuit.
(*k*) Differential pressure gauge across each boiler.
(*l*) Outside temperature remote-thermometer.
(*m*) Coal meters on the automatic stokers.

Most of the above may be combined with recorders to give a continuous chart of the conditions obtaining, though this additional expense is rarely justified.

Another instrument which is sometimes provided, and which is in any case useful for keeping an accurate check on operating costs, is a *B.T.U. Meter*. This is rather expensive, and generally only justified in cases where costs have to be allocated over various parts of the system.

The following description of one modern B.T.U. Meter is taken from a paper* by Dr. F. M. H. Taylor:

'The differential type of flow meter consists of a differential pressure producing unit inserted in the pipe line. This may be in the form of a thin orifice plate, a nozzle, a short or long Venturi tube. The differential pressure caused by such devices operates the instrument which may be

* 'Measurement of Heat Output in Hot-water Circulating Systems', by F. M. H. Taylor, Ph.D., *Journal of the I.H.V.E.*, Vol. 9, 81.

described as a manometer, equipped with a float-operated mechanism so arranged that the indications may be accurately calibrated in terms of water flow.

'The pressure loss caused by the orifice plate, or other differential producing medium, need not exceed, at maximum calibration of the instrument, some 70 to 80 inches of water. Under normal flow conditions, a pressure loss of half this figure would be anticipated. Furthermore, the use of a Venturi tube in place of the orifice plate will reduce the pressure loss considerably, and, in fact, by careful design and choice of the proper type of flow measuring instrument, it is possible to arrange that the pressure loss is negligible when compared with the loss through the system as a whole.

'For the measurement of temperature difference, consideration was given to the use of various types of temperature measuring equipment, and, for reasons of accuracy and co-ordination, electric temperature measurement was finally used.

'In order to provide an instrument which will read in terms of B.T.U. it is necessary to measure the weight of flow in pounds and to multiply this figure by the temperature rise or drop in the system.

'Calibration of the two factors of temperature difference measurement and of water flow to give an integrated B.T.U. quantity, is achieved by equipping the flow meter with integrating mechanism and so arranging that, after the measurement of a predetermined weight of water, an electrical impulse is given to cause operation of a second integrator, calibrated in B.T.U., which is also influenced electrically by the temperature difference in the flow and return mains.'

SAFETY DEVICES

If the pressure in a H.P.H.W. system were suddenly reduced, as for instance by the bursting of a pipe, large quantities of steam would be generated by the expansion of the superheated water. Further, the circulating pumps would continue to feed water into the system, with grave risk of lowering to a dangerous extent the water-level in the boilers.

Automatic Isolating Valves—To protect the system against such an occurrence, automatic isolating valves are sometimes provided in the mains where they leave the boilerhouse. Fig. 212 shows a cross-section of one such valve, consisting of a steel ball normally resting on a cup in the lower part of the casing. If, however, the velocity of the water rises above a predetermined value, the ball is lifted and forced into the seating, thus closing the valve.

The operating point can be altered within limits by raising or lowering the cup which carries the ball. A bye-pass is provided round the valve to equalize the pressure when required, so that the ball is released, and falls back to its normal position. One such valve is fixed in the flow and one in the return main, the direction of course being such as to prevent the flow of water away from the boiler.

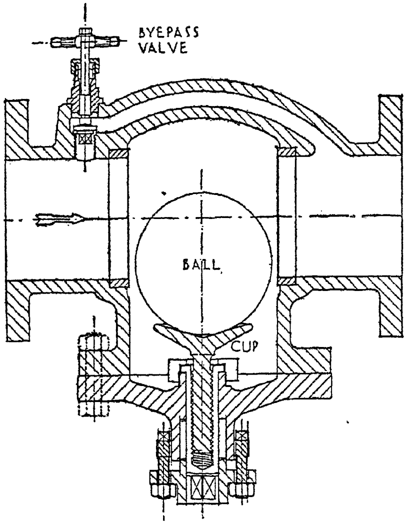


FIG. 212.—Automatic Isolating Valve (Hobdell).

The immense hydraulic force behind such a valve on closing is considered by some engineers to be dangerous, and there is for this reason a tendency to omit them as accidents though broken pipes appear to be rare.

'Low Flow' Devices—Reference has already been made to the necessity of maintaining a certain minimum rate of flow through the boilers to prevent overheating of the tubes and generation of steam. Such a condition may arise due to excessive bye-passing at the mixing point, or it may be caused by any of the following:

Tubes or piping partially blocked with sediment, etc.
One of the boiler stop valves not fully open.
Only one circulating pump being run instead of two.

A much worse condition, of course, arises if there is complete closing of the valve or if the pumps stop altogether. The stoppage of a pump is more serious with a tubular boiler than with a shell-type.

Fig. 213 shows diagrammatically one type of device to give warning of low flow through a boiler. It consists of an orifice plate inserted in the connection to the boiler, the drop in pressure across this orifice varying with the flow in the pipe. The pressure drop is transmitted to any convenient instrument for operating an alarm, or setting in motion any sequence of operations such as the stopping of the draught-fans and automatic stokers.

Other Safety Devices—A great variety of other alarms and protective devices may be devised for the system, and while such apparatus no doubt serves a useful purpose if maintained in perfect working order, it is the

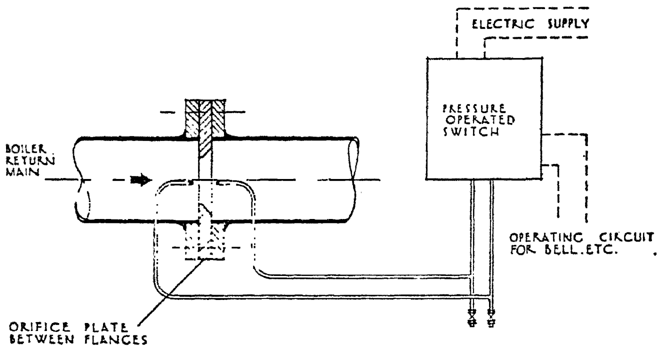


FIG. 213.—Diagram of Boiler Low-flow Alarm Device.

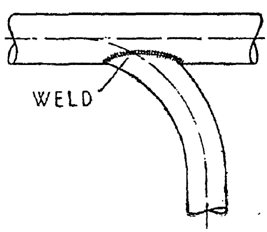
authors' experience that the provision of safety devices can easily be overdone, and that if not properly maintained they constitute a danger, since the operating staff rely upon them and do not perhaps pay such close attention to the working of the system as they might otherwise do, so that the failure of an alarm is doubly serious.

Protective devices if installed should be kept as simple as possible, and so designed that their failure, or the failure of the electrical supply to them, does not preclude quick action in an emergency. In any case, provision must be made for periodical testing and for cutting out the whole device if found faulty.

PIPING FOR H.P.H.W. SYSTEMS

Piping is invariably of mild steel, suitable for the pressure. In this connection it must be remembered that at certain points of the system the pressure is in excess of the boiler working-pressure by an amount equal to the pump head, i.e. a system with boilers at 100 lbs. per sq. inch and a pump head of 90 ft. total would have to be designed for approximately 140 lbs./sq. in.

Joints—Screwed joints are not suitable for H.P.H.W. piping, and should never be used, as they are liable to leak in time. All valves, etc., should be flanged, and elsewhere the piping should be butt-welded, either by oxy-acetylene or by electric arc.



Welding, if properly carried out by experienced welders, will give joints as strong as the piping itself, and it would appear unnecessary to provide stiffeners or bracing to reinforce the joints, though this is sometimes done on larger sizes.

All bends should be of large radius (at least 3 pipe diams.), and tees should be 'swept' so as to avoid sudden drops of pressure in the piping and with the danger of air and steam collecting.

Connections—Connections to heating apparatus are best made with flanges, but to save expense the piping is sometimes welded direct to a 'tail' left on the heater battery, etc. This is quite justifiable, since welding technique has now developed to a point where it is as easy to cut and reweld as it is to break and remake a flanged joint.

Headers and branches in the boilerhouse may also be built up from standard tubing and bends welded together, but it is desirable to provide a certain number of flanges in the boilerhouse to facilitate erection and dismantling. Connections to pumps and boilers will generally be flanged.

Expansion of the Piping is a matter of the first importance, and should be carefully considered at every stage of the design. Short rigid connections between boilers, drums, headers, pumps, etc., should be avoided, and it is worth pointing out that whereas such a short connection might be practically unstrained if all the piping adjacent to it is full of hot water, cases will arise when part of the piping is out of use, and consequently full of cold water, with resulting uneven contraction, and strain.

The mains outside the boilerhouse will behave as regards expansion in exactly the same way as steam mains at the same temperature, and the basic considerations are the same as those already discussed in Chapter XIV. Some typical H.P.H.W. mains will be seen in Plate XXIV (facing p. 365).

Air Bottles—All high points of piping should be provided with vent pipes taken off the top of the main and run to an air bottle, as shown in Fig. 214 (*a*). The vent pipe should be of at least 1 in. diameter for small pipes, or 1¼ in. for pipes over 6 in. bore, and if not vertical should rise to the air bottle. This is made up from a short length of

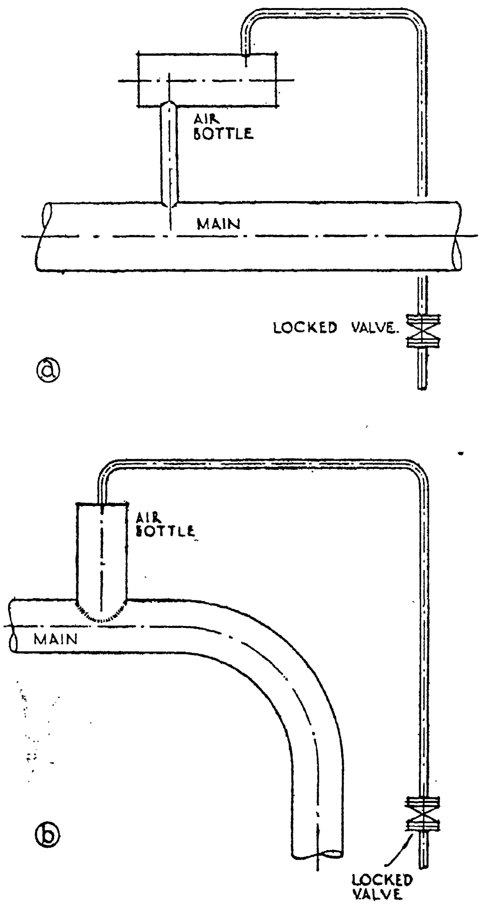


FIG. 214.—Air Bottles on Mains.

pipe about 6 in. bore, and has an air-pipe taken from the top to a convenient level and provided with a stop-cock, which should always be locked in some way to prevent unauthorized use.

Alternatively, a pocket may be formed on top of the pipe as in Fig. 214 (*b*), and an air-pipe taken off the top as before. This has the advantage that air bubbles carried along the top of the main by the water have a greater chance of being trapped in the large diameter pocket than of entering the relatively small vent pipe in (*a*).

Venting—Mains should be fixed to a slight fall to assist in venting, and ideally all mains should rise in the direction of flow so that the entrained air does not have to collect against the flow of water. This, however, would entail fixing the mains with falls in different directions, which is inconvenient, and is not generally necessary. The main which would be expected to contain the greater quantity of air (normally the flow main) should rise in the direction of flow, with the return main lying parallel to it.

Once the initial air has been removed there is generally little further accumulation. Due to the considerable pressure in the system, air is compressed to a much smaller volume than in the normal low-pressure system.

Valves—Trouble is often experienced in H.P.H.W. systems due to leakage past the glands of the valves. As soon as it is at atmospheric pressure, the water leaking out immediately evaporates and leaves behind any salts which it may contain, and this deposit may in time, if not regularly removed, be sufficient to prevent the valve from being turned. Trouble is particularly noticeable with small motorised valves where the operating torque is limited to the power of the motor. To overcome this difficulty many attempts have been made to design special glands for H.P.H.W., and this point should be borne in mind when specifying makes and types.

Valves used for H.P.H.W. should always be flanged, and any good quality full-way steam valve with special gland and packing will give reasonably satisfactory service.

The cost of providing flanged valves and counter-flanges for even the smallest connections, such as those to unit heaters, has led valve manufacturers to produce various alternative designs, one of which uses a special type of screwed union for connecting the valve body to a tail, which is then welded to the piping.

Regulating Valves—The amount of water passing through a H.P.H.W. system is relatively small, and the pump head available at the nearer branches may be large, so that regulation is often critical. Some designers, therefore, provide, in addition to the isolating valves on the flow and return to the branch, a third valve in the return which is used for regulating only, and once set this valve need never be disturbed. This method is somewhat expensive, and recently combined regulating and shut-off valves have been produced. The regulation is effected by means of an auxiliary seating, which is so arranged that it cannot be tampered with after adjustment.

Valves in the boilerhouse and important valves throughout the system should be of high quality parallel-slide type, and a little extra expenditure on this item should save trouble later on.

Starting-up—Starting a system up from cold should be done very gradually, so that the piping has time to adjust itself to the expansion. The procedure is to start-up the whole system from cold, and gradually warm up the water as it circulates. Alternatively, the boilers may be warmed up and raised to a moderate pressure with closed valves, and the return valves slowly opened so as to apply the static pressure to the system. The pump is then started against closed valves, and the cold water set into motion gradually by opening the valves, including the mixing valve. Next, the boiler flow valves are opened and the water and piping gradually warmed up, and lastly the mixing valve may be closed down as required, to achieve the required working conditions.

HEATING APPARATUS

The heat may be utilized in the building by means of convectors, unit heaters, and pipe-coils. These follow very closely the designs already described for steam in Chapter XIV, with the exception that the flow and return connections are of the same size, and no steam traps are used.

The discussion on final air temperatures, fresh air inlets, layout, thermostatic control, and so on (see pp. 322-324), applies equally to steam and H.P.H.W. unit heaters, and need not be repeated here.

H.P.H.W. is also used as the heating medium for air-heater batteries for plenum and ventilation systems, as discussed in Chapter XVII.

Plate XXII (facing p. 361) shows the upper part of a large shed heated by H.P.H.W. The vertical discharge unit heater, the mains (not lagged when the photograph was taken), main isolating valves to the branches, unit heater connections and valves, and a pair of air-pipes with valves, will be seen on the left-hand side of the plate.

Emission from Heating Surfaces—The heat-loss from uninsulated piping is considerable, and it should be remembered that it occurs from all flow and return mains and connections. All external piping should be very well insulated, also the larger internal piping, especially where serving a long building or a series of buildings, as otherwise it will be impossible to maintain an adequate water temperature at the far end. Connections to unit heaters and convectors, however, may be left unlagged, and the heat emitted reckoned as useful. This not only saves lagging, but reduces, sometimes to a surprising extent, the size of the units.

Table LXIII (p. 382) gives the emission from bare pipes to air at normal temperatures. The Table also gives the emission from convectors, for the working temperatures stated.

It will be seen that the emission from a convector does not depend entirely on the mean water temperature. Actually it has been found to depend largely on the velocity of the water through the battery, which

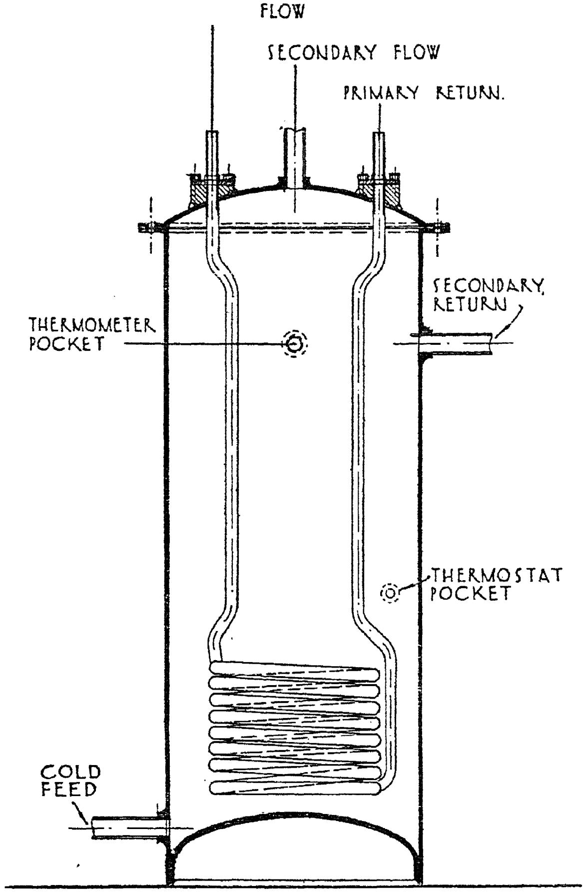


FIG. 215.—H.P.H.W. Vertical Storage Calorifier for Hot-water Supply.

velocity, for a given battery, is a function of the temperature drop across it.

Hot-Water Supply—For domestic hot water the H.P.H.W. serves as the primary supply to a water-water calorifier, as shown in Fig. 215. The heating coil is a continuous helix, so as to maintain a high velocity, and is removed from the top by unbolting the domed cover to which the coil is welded. Provision is made for an on-off or direct-acting thermostat controlling a thermostatic valve in the primary line, either flow or return.

There will also be required:

Thermometer showing secondary water temperature.
Drain cock.
Cold water feed.
Open vent pipe on secondary flow.

TABLE LXIII

(*a*) EMISSION FROM BARE PIPES IN B.T.U.'S/FT. RUN/HOUR TO AIR AT 50°–70°

Nominal Pipe Size	Temperature Difference in °F. between Pipe and Surrounding Air						
	150	175	200	225	250	300	350
½"	95	117	139	166	192	251	319
¾"	115	141	171	199	233	304	385
1"	140	172	206	243	284	372	472
1¼"	170	210	251	297	347	455	578
1½"	191	235	282	334	389	510	650
2"	235	288	346	410	478	628	801
2½"	276	340	408	483	563	742	947
3"	330	406	488	578	674	888	1134
4"	413	510	622	726	849	1140	1428
5"	494	609	733	869	1015	1339	1710
6"	573	707	850	1009	1180	1557	1992
8"	732	904	1088	1289	1510	2000	2555
10"	886	1092	1315	1560	1825	2415	3100
12"	1033	1279	1537	1830	2140	2840	3635

Based on *I.H.V.E. Guide.*

(*b*) EMISSION FROM CONVECTORS OF 'VECTAIR' TYPE TO AIR AT NORMAL ROOM TEMPERATURE

Water Temperature °F. Flow	Return	Emission in B.T.U.'s/sq. ft./hour
360	280	250
340	300	295
330	310	330
340	260	230
320	280	270
310	290	305
320	240	210
300	260	250
290	270	280

Information from Messrs. British Trane Co. Ltd., for 'Vectairs' having ½" steel tube elements with copper fins.

Table LXIV gives the transmission coefficients recommended by the *I.H.V.E. Guide*, for water-water calorifiers, including the allowance for furring of the coil in the case of hard water supplies.

Fig. 216 shows a typical arrangement using such a calorifier, for hot-water supply to a works ablution-block. The secondary pipe sizing will follow the methods given in Chap. XI.

Hot-Water Heating—For a building or group of buildings where H.P.H.W. heating is not suitable, a low-temperature radiator system may be installed, and served from a water-water calorifier using H.P.H.W. for the primary heating.

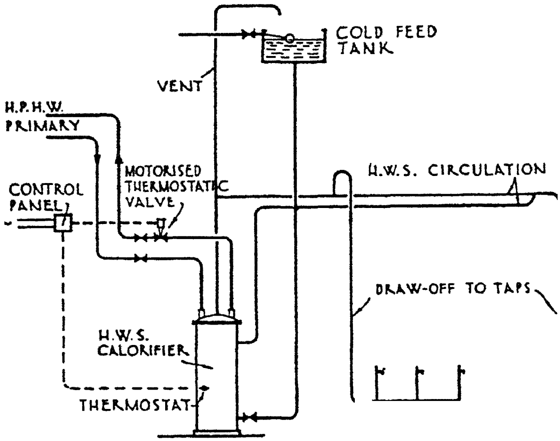


FIG. 216.—Typical arrangement of Local Hot-water Supply System served from H.P.H.W.

TABLE LXIV

WATER-WATER CALORIFIERS

Heat transmission in B.T.U.'s/hr. per °F. difference between mean primary and mean secondary, per sq. foot of external tube surface. Tubes 1″–1½″ diameter spaced not less than one diameter apart.

Mean Primary Temperature °F.	Mean Secondary Temperature °F.	Water Velocity in Tubes ½ ft./sec. Soft Water	½ ft./sec. Hard Water	1 ft./sec. Soft Water	1 ft./sec. Hard Water
200	50	70	61	90	73
	100	72	63	91	74
	150	74	65	92	75
250	50	83	70	109	87
	100	87	75	114	91
	150	93	79	118	95
300	50	104	88	133	103
	100	111	93	138	107
	150	116	99	147	114

Based on *I.H.V.E. Guide.*

The calorifier would be similar to that shown in Fig. 215, except that it would be of non-storage type. The mountings and method of sizing would also be similar, but there is, of course, no need to allow for furring, since the secondary water in this case is not drawn off.

Fig. 217 is a diagram of the piping and apparatus for the heating and ventilation of the office-block in a factory heated by H.P.H.W. The incoming H.P.H.W. mains are connected to headers which in turn serve the heating calorifier, with its own low-pressure radiator system, pump-circulated.

In addition, connections will be seen from the H.P.H.W. headers to H.W.S. units and plenum air-heaters.

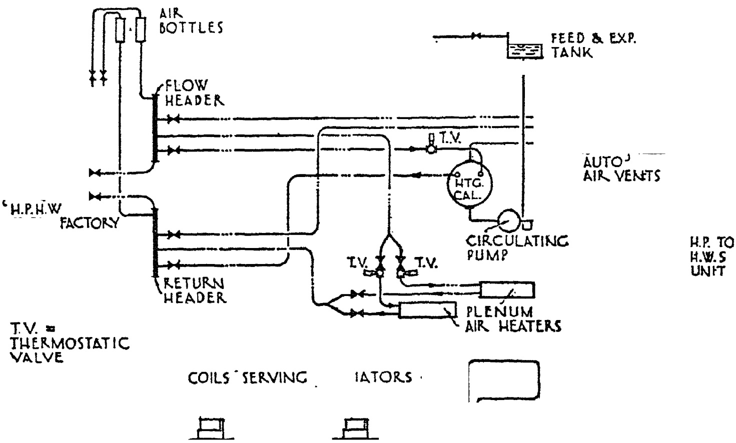


FIG. 217.—Diagram of Radiator System served from H.P.H.W. (see text).

The design of the low-pressure system will be exactly as given in Chapter IX. In sizing the calorifier a suitable margin should be added to the net heat losses to allow for time-lag and heating-up (see pp. 64-66).

Other Applications—H.P.H.W. may be used as a source of heat for many sorts of apparatus, among which the following may be mentioned.

Steam Generators, which are in effect water-steam calorifiers, very similar to the H.W.S. unit already described, and using a primary coil of the same type.

Cooking Apparatus, hot closets, carving tables, etc., have in the past usually been supplied with low-pressure steam from a low-pressure steam generator, but as H.P.H.W. becomes more common, specially designed apparatus capable of using the hot water direct, should become available.

Vats and Boiling Vessels of all types. In these a coil is placed in the bottom of the vessel, or in a separate chamber at the side with mechanical circulation.

Baking and Enamelling Ovens, and similar apparatus requiring an intense radiant heat, and hot air for drying.

Vulcanizing and Plastic Presses where hot surfaces are required to promote a physical or chemical process.

Laundry Dryers, etc., in which hot dry air is used. The air is circulated over the H.P.H.W. coil by means of a fan.

The details of the design of such apparatus are beyond the scope of the present book, but it will be seen that all the processes normally served from a steam-heating system may equally well be provided by H.P.H.W. with the added advantage of greater range of control by valve operation.

PIPE SIZING FOR H.P.H.W. SYSTEMS

The principles and method are exactly the same as those already explained for forced-circulation low-pressure systems (see pp. 206-209). At the temperatures of working for H.P.H.W., however, the properties of water are different from those assumed in constructing Table XXXIX, which is for water at 140°-180° F. Thus, the viscosity of water at 300° is approximately half of that at 160°, and the density is less, as will be seen from Table LXI. Table LXV has therefore been compiled, and corresponds in every way to Table XXXIX, except that it is based on a water temperature of 300° F. The parts of the latter table applicable to natural circulation have, of course, no counterpart in Table LXV.

Temperature Correction Factors—While the earlier table was sufficiently accurate for all temperatures normally encountered in low-pressure systems, in H.P.H.W. the temperature range is much wider and no one table will apply throughout. Assuming that any variation of head within ±2 per cent. may be ignored, Fig. 218 shows the percentage corrections outside this limit which must be applied to the resistance heads as found from Table LXV to give the correct heads at 200°, 250° and 350° F. respectively. Thus, to find the resistance of 200 ft. run of 4 in. main carrying 70,300 lbs./hour at 350° F. :

From Table LXV, at a velocity of 48 in./sec.	h/ft. = 0·12 in./ft.
For this temperature, velocity and pipe-size, from Fig. 218, correction	= +2·6 per cent.
∴ Resistance head required	= 1·026 × 0·12 × 200
	= 24·6 inches w.g.

In theory a different correction factor should be applied to the resistance of each length of piping separately, in accordance with its size and the velocity of flow through it. In practice this can be approximated as described later.

Outline of Method—To size the external piping for a H.P.H.W. system serving a number of buildings: Design flow and return temperatures 300° and 200° respectively.

Having a site plan with the runs of the mains and net heating loads marked for each building, proceed as follows:

(a) Total heat of water at 300° - - - - - - -	*269·7 B.T.U.'s/lb.
" " " 200° - - - - - - -	*168·1
Heat carried per lb. - - - - - -	101·6 B.T.U.'s.

* Interpolated from Table LX.

Quantity of Water in Lbs. per Hour for Various Pipe Sizes.

	Friction Loss in Inches W.G. per Foot	½"	¾"	1"	1¼"	1½"	2"	2½"	3"	3½"	4"	5"	6"	7"	8"	9"	10"	12"	
		V=6"/sec				V=12"/sec					V=24"/sec				V=36"/sec				
	·03	150	420	910	1610	2620	5600	10000	15800	23000	34000	58500	94,000	145000	215000	280,000	380,000	610,000	V=48"/sec
V=6"/sec	·04	175	490	1050	1890	3050	6350	11500	18000	27000	39000	66000	110,000	170,000	245,000	330,000	440,000	710,000	
	·05	190	550	1190	2120	3400	7200	12800	20500	30000	45000	76000	123,000	190,000	275,000	370,000	500,000	800,000	
	·06	210	600	1260	2350	3800	7800	14200	22500	33000	49000	82500	136,000	212,000	300,000	405,000	555,000	900,000	
	·07	230	650	1400	2500	4000	8500	15500	24000	36000	53500	90000	148000	230,000	330,000	440,000	600,000	970,000	
	·08	250	700	1500	2700	4350	9100	16600	26000	39000	56500	97000	158,000	240,000	360,000	475,000	640,000	1,050,000	V=72"/sec
	·09	260	740	1600	2850	4600	9700	17500	27500	41500	60,000	101500	170,000	260,000	375,000	500,000	680,000	1,100,000	
	·10	280	780	1670	3000	4800	10300	18800	29800	43000	65000	110,000	180,000	275,000	400,000	530,000	715,000	1,160,000	
	·12	300	860	1820	3280	5300	11300	20700	32000	48000	70300	120,000	198,000	305,000	440,000	590,000	800,000	1,250,000	V=96"/sec
V=12"/sec	·14	320	940	2000	3600	5700	12300	22400	35000	53000	76000	130,000	215,000	325000	480,000	630,000	870,000		
	·16	350	1000	2150	3800	6200	13200	24000	37500	57000	83000	140,000	230,000	350,000	510,000	680000			
	·18	375	1080	2280	4100	6600	140,00	25700	40000	60000	88000	150,000	245,000	375000	540,000	720,000			
	·20	395	1120	2400	4300	6900	14800	27000	43000	63000	94000	160,000	260,000	400,000	580,000				
	·23	420	1200	2600	4600	7450	15800	29000	45000	69000	100,000	170,000	280,000	430,000	630,000				
	·26	450	1300	2750	4950	8050	17000	31000	48000	72500	108000	180,000	300,000	450,000					
	·30	490	1400	3000	5250	8700	18000	33500	52000	78000	115,000	195,000	325,000						
	·35	525	1500	3250	5750	9300	19500	35500	56500	85000	123000	210,000	350,000						
	·40	560	1610	3450	6150	9800	21000	38000	61000	91500	135,000	227,00							
V=24"/sec	·45	600	1700	3550	6500	10600	22700	41500	65000	97500	140,000								
	·50	615	1810	3850	6950	11100	23500	44000	68000	102000	150,000								
	·60	680	2000	4200	7600	12200	27000	47500	76000	112,000									
	·70	760	2180	4550	8300	13200	28000	52000	81000										
	·80	800	2300	4000	8800	14100	30000	55000											
V=36"/sec	·90	850	2450	5200	9400	15000	32500	59000											
	1·00	900	2600	5500	9900	16000	34000	62000											
	1·20	1000	2850	6000	10900	17500	37000												
	1·40	1080	3120	6500	11800	19000													
V=48"/sec	1·60	1130	3300	7000	12500	20500													
	1·80	1200	3550	7400	13200														
	2·00	1280	3750	7900	14000														
	2·50	1410	4150	8700	15600														
	3·00	1550	4550	9700															
		V=72"/sec.	V=36"/sec																

FOR WATER AT 300° F.

LOCAL RESISTANCES.

FULLWAY VALVE	R = 0·5	BOILER.	ADD FEET HEAD SEPARATELY, AS GIVEN BY MAKERS.
GLOBE VALVE	R = 2·0	UNIT HEATER	
BEND SHORT RADIUS.	R = 2·0	CONVECTOR	
" LONG RADIUS	R = 1·0.	CALORIFIER	
TEE SWEEP	R = 1·0	ETC.	
" RT. ANGLE	R = 3·0.		

	Nominal Bore of Pipe in Inches																
V.	½"	¾"	1"	1¼"	1½"	2"	2½"	3"	3½"	4"	5"	6"	7"	8"	9"	10"	12"
6·0"	1·2																
12·0"	1·3	2·0	3·0	4·0	5·0												
24·0"	1·4	2·2	3·5	4·5	5·5	7·7	10·5	13·5	16·0	19·0							
36·0"	1·6	2·3	4·0	5·0	6·0	8·0	11·0	14·0	17·0	20·0	23·0	32·0	38·0	46·0	55·0		
48·0"	1·7	2·6	4·0	5·2	6·5	8·7	12·0	15·0	18·0	22·0	25·0	34·0	41·0	50·0	60·0	68·0	86·0
72·0"		3·0	4·0	5·5	7·0	9·5	13·0	16·0	19·0	25·0	28·0	36·0	43·0	54·0	65·0	72·0	93·0
96·0"			4·3	5·8	7·8	9·8	13·5	17·0	21·0	25·0	29·0	38·0	44·0	57·0	68·0	78·0	98·0

EQUIVALENT RESISTANCE OF R = 1 IN FEET OF PIPE.

TABLE LXV

Flow of Water in Pipes at 300° F.

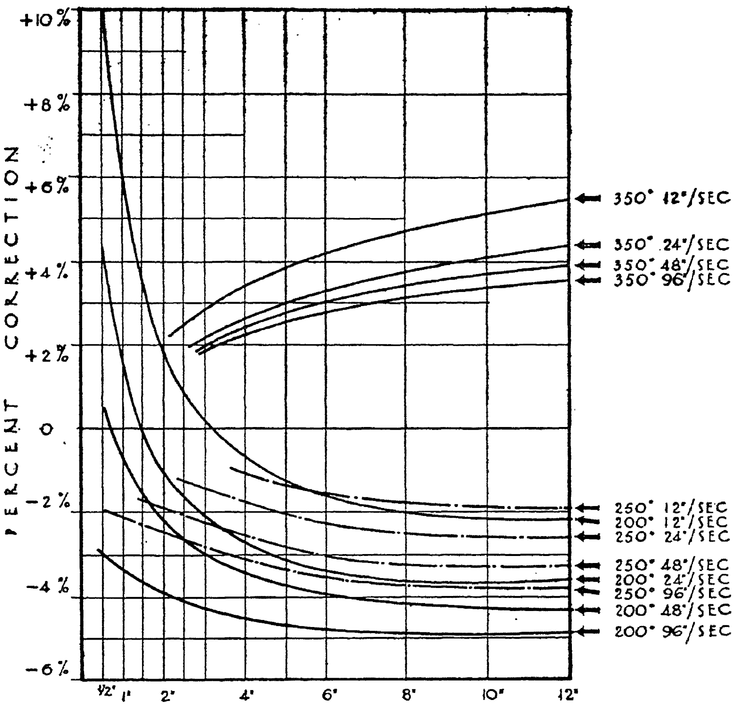


FIG. 218.—Corrections for Temperature, to be applied to Resistance Heads found from Table LXV.

(*b*) As a first approximation assume a mains heat loss of 5 per cent. Then 'H' B.T.U.'s *net* corresponds to $\dfrac{H \times 1{\cdot}05}{101{\cdot}6} = {\cdot}01035H$ lbs./hr. *gross*.

(*c*) Add back the heating loads along the main to the boilerhouse, applying the factor in (*b*) in each case.

(*d*) From Table LXV find the pipe-sizes for each load, for a resistance of 0·08–0·14 inch/ft. subject to the velocity not exceeding 4–5 ft.

(*e*) Before going any further, the pump head should be taken out roughly for the sizes thus obtained, and using equivalent runs measured off the drawing, to make sure that the pump head is not excessive.

(*f*) The losses from the lagged mains at the known temperatures may now be calculated, and the actual losses added into the heat-loss totals in place of the 5 per cent. assumed in (*b*). The mains-losses will be proportioned in the ratio of the heating loads in the manner given on p. 203.

(*g*) Now calculate the pump head from Table LXV for the known pipe-layout, allowing in the equivalent runs for the bends and local resistances in accordance with the lower part of the table. The flow and return resistances should be kept separate.

(*h*) The flow-main resistance need not be corrected for temperature, as the table is based on 300°, the actual flow temperature. The return-main resistance calculation should be analysed to find the velocities and pipe sizes. It will probably be found that by far the greatest part of the resistance is due to pipes (for example) 5″–8″ in size, and with velocities round 3–5 fs. To a sufficient degree of accuracy, therefore, an average correction of −4 per cent. (see Fig. 218 for 200°) may be applied to the whole of the return resistance. A small margin should be allowed in the pump head as explained previously.

(*i*) From the resistance calculations mark on the drawing the residual head left at each main branch. The branches should then be sized for a greater drop per foot than was used for the index circulation, so as to absorb as much of the residual head as possible. This balancing of the system is essential with H.P.H.W. where the amount of water circulating is relatively small.

(*j*) Small branches and connections to apparatus may be sized from the table and Fig. 218 in the usual manner.

Losses from Insulated Piping—The importance of good insulation has already been stressed, and a definite standard should be laid down by the designer.

The efficiency of lagging varies with the temperature difference between the pipe surface and the outside air, so that the data given in Table XLI (p. 218) for 100° F. difference will not apply to H.P.H.W. piping. Table LXVI gives the heat emissions from insulated pipes for the range of temperatures applying to H.P.H.W. The figures for 85 per cent. magnesia have been taken from the *I.H.V.E. Guide*, and a further column added giving to the nearest eighth of an inch the corresponding glass-silk thickness, to give approximately the same insulation efficiency. The table should be used in conjunction with Table LXIII (*a*) to obtain the actual emission.

It will be seen from the table that the percentage emission for any given pipe varies with the temperature difference, and also that the thickness of glass-silk corresponding to 1 in. and 2 in. of magnesia varies slightly over the temperature range. This difference in characteristic of the two materials is more pronounced at higher temperatures. Thus at 700° difference it would require approximately $1\frac{3}{4}$ in. glass-silk to give the same efficiency as 2 in. magnesia on a 4 in. pipe.

TABLE LXVI

HEAT EMISSION FROM INSULATED PIPES, EXPRESSED AS A PERCENTAGE OF THE EMISSION FROM BARE PIPES, TO STILL AIR AT NORMAL ROOM TEMPERATURE.

Temperature diff. between Air and Water, °F.	Thickness of Magnesia*	Approx.† corresponding Thickness of Glass-silk	Nominal Pipe Sizes					
			$\frac{1}{2}''$	1″	$1\frac{1}{2}''$	$2\frac{1}{2}''$	5″	10″
150	1″	$\frac{5}{8}''$	25	22	20	19	18	17
	2″	$1\frac{1}{8}''$	17	16	14	13	11	11
200	1″	$\frac{5}{8}''$	23	21	19	18	16	16
	2″	$1\frac{1}{4}''$	17	15	14	12	10	10
250	1″	$\frac{5}{8}''$	22	20	18	17	15	15
	2″	$1\frac{1}{4}''$	17	15	13	12	10	9
300	1″	$\frac{3}{4}''$	20	18	16	16	14	14
	2″	$1\frac{3}{8}''$	16	14	12	11	9	8

* Consisting of 85 per cent. magnesia and 10–15 per cent. asbestos fibre. Effect of hard-setting finishing coat has been ignored.

† See note in text.

Table based on *I.H.V.E. Guide*, with column 3 added.

SPECIAL SYSTEMS

The foregoing descriptions of H.P.H.W. systems are intended to cover the main principles. There are, however, several 'patented' systems which employ special apparatus or piping arrangement. The special features mostly deal with the method of mixing, the balancing of the system, and the arrangement of take-off at the boilers, coupled with special designs of heat-emission and heat-exchange apparatus, valves, etc. Such special systems need not be described here, as no one should attempt to install a H.P.H.W. system who has had no previous experience of it, and the details might only be confusing to those who are not familiar with the finer points which require attention.

CHAPTER XVI

Running Costs of Heating Systems

The type of fuel for a heating system will be determined by cost, cleanliness, convenience and availability.

Cost depends on quantity of heat required and price per unit quantity. The amount of heat required for any given plant and conditions can be arrived at with a fair degree of accuracy, but will naturally vary from year to year according to the weather. Efficiency and ratio of period of utilization to total hours will affect the comparison as between one fuel and another. The price per unit quantity of heat will generally be the determining factor, and this is controlled by the charge per therm of gas, or per unit of electricity, or, in the case of solid and liquid fuels, per ton of coal, coke or oil, and their calorific values.

With gas and electricity the heat delivered per therm or per unit is fixed, but not so with other fuels. Thus, in order that coal, coke or oil may be charged for on their heating value it has often been suggested, as indeed in some large contracts is already done, that fuels be priced on a calorific basis. There is much in favour of this course, and its adoption may some day become general. Thus, instead of ordering 1 ton of coal from the merchant, he will be asked to deliver, say, 250 therms of coal.

Cost must include other items, and is made up of

(1) Fuel consumption (referred to above).
(2) Running of auxiliaries.
(3) Labour.
(4) Maintenance and repairs.
(5) Interest and depreciation.
(6) Insurance.

These will be dealt with in detail later.

Cleanliness and convenience are equally important, yet it is often difficult to give a money value to them. Conditions of cleanliness which might be quite satisfactory, for instance, in a factory or workshop would probably be intolerable in a bank or office building. In addition, owners of two buildings of exactly similar use might have widely different ideas of a minimum standard of cleanliness and convenience.

The fuels and methods of firing come in the following order, the least cleanly and convenient being placed first:

Hand-fired coke or coal.
Hand-fired anthracite.
Automatic coal stokers.

Magazine-fired boilers with top feed.
Oil firing.
Gas firing.
Electric thermal storage.
Electric direct heating.

As to convenience, nothing can equal the turning on or off of a switch, possible only with electric systems, or the turning of a cock with gas. The question is whether owners are prepared to concede an increase in running cost on account of these advantages, and if so, how much.

The availability of fuel has not troubled us in the past. Most types have been obtainable if required. The future may bring a National Fuel Policy into being, and the choice might then be controlled by best use of fuel resources, reduction of transport, etc. Presumably costs would also have to be adjusted on a national scale. Such a development is at the moment conjectural, and its implications cannot be discussed further.

1. FUEL CONSUMPTION

The quantity of fuel required for the warming of a building depends on

(*a*) heat input to maintain the specified temperature;
(*b*) calorific value of fuel;
(*c*) average working efficiency over the season;
(*d*) period of use;
(*e*) proportion of full load operation.

(*a*) **The Heat Input** is the basis of any calculation of fuel consumption. This depends on the heat losses, which can be calculated and must include any losses from mains. In fact, the heat input figure must cover the maximum transmission from all the heating surfaces and piping, based on 30° or 32° F. outside temperature.

Incidentally, direct electric and gas systems which have no mains losses score here by starting with a lower initial heat requirement.

(*b*) **Calorific Value** is a clearly established figure for each fuel, and has already been discussed. It must be remembered that solid fuels often vary as a result of the amount of moisture which they contain. Moisture may vary from 5 to 10 per cent. of the weight, being often very high in coke.

A further point is that where coal is being bought direct from a colliery the colliery weight has to be accepted. This can rarely be made to agree with the weight received in the boilerhouse, the difference generally being on the wrong side.

Thus, with solid fuels some allowance should be made for these factors by assuming a definite loss of, say, 10 per cent. by weight.

(*c*) **Average Working Efficiency over the Period**—This is the most controversial point of all, since test figures can be produced which, while probably true for the conditions under which the test was made, are very misleading for ordinary working.

It has already been stated that small hand-fired boilers may be no more than 40 per cent. efficient and that 50 per cent. is a fair figure with normally unintelligent firing. It has also been stated that the application of modern thermostatic controls to a hand-fired boiler may reduce fuel consumption by as much as 20 per cent., this being partly due to the better combustion assured thereby, as well as by the reduction of overheating. This has been demonstrated many times in practice and is not a theoretical figure.

The authors therefore consider it only fair, when hand-fired solid fuel is being considered, that it should be on the assumption that the boiler is fitted with up-to-date controls, in order to be comparable with other methods of firing which would almost certainly have the same advantage. With this in mind it is felt that an overall working efficiency for the season of 60 per cent. fairly represents what may be expected with present-day apparatus and reasonably intelligent supervision.

In gravity feed magazine coke boilers with automatic controls a figure of 75 per cent. should be maintained indefinitely if the flues and tubes are kept clean, and ash, etc., regularly removed.

Automatic stokers and oil firing depend largely, so far as their efficiency is concerned, on the regulation of the air supply and on the type of boiler to which they are connected. It has already been stated that lack of adjustment may reduce the efficiency to 50 per cent. or less, but here again any comparison must be on the assumption that the system is modern and properly maintained, and when this is so the figures may be 70 per cent. with automatic stokers and 75 per cent. with oil. The authors know of several examples where records show that these efficiencies are maintained year in and year out.

Gas-fired boilers may be taken at 85 per cent. without much argument and electric thermal storage systems at 98 per cent.

TABLE LXVII

(*a*) Efficiency of Installations with Different Fuels and Methods of Firing
(All assumed to have Full Thermostatic Control)

Fuel and Method of Firing	Efficiency	Potential B.T.U. per Lb. (or Unit)	Output in B.T.U.'s per Lb. (or unit) at Efficiency Given	Therms per Ton	With Deduction of 10 Per Cent. for Sundry Losses (Solid Fuel) Therms per Ton
	Per Cent.				
Gas Coke, hand-fired	60	12,000	7,200	161	145
Furnace Coke or Coal, hand-fired	60	13,000	7,800	174	157
Gas Coke, Magazine Boilers	75	12,000	9,000	202	182
Anthracite, hand-fired	60	14,000	8,400	188	170
Coal, with Automatic Stoker	70	13,000	9,100	204	184
Oil Fuel	75	18,000	13,500	302 therms per ton.	
Gas (in Boiler)	85	100,000	85,000	0·85 therms per therm.	
Electricity (Thermal Storage)	98	3,415	3,350	0·0335 therms per unit.	

TABLE LXVII—*continued*

(*b*) COST PER THERM FOR VARIOUS FUELS

Fuel	Cost*	Cost per Therm at 100 Per Cent. Efficiency (from Table XV, p. 51)	Cost per Therm from Table (*a*) above
		d.	d.
Electricity	¼d. per unit	7·3	7·5
Gas	9d. per therm	9·0	10·6
	4d. per therm	4·0	4·7
Fuel Oil	80s. per ton	2·4	3·2
Coal or Coke	60s. per ton	2·7	5·0
(60 per cent. efficiency)	40s. per ton	1·8	3·3
12,000 B.T.U.'s/lb.	30s. per ton	1·35	2·5
	20s. per ton	0·9	1·65

* These costs must, of course, be revised in the light of actual prices. See note in Preface.

With the above factors, Table LXVII has been constructed, deducing the net therms per ton obtainable from various fuels fired in boilers under the conditions stated.

As a matter of interest the corresponding costs per therm have been taken out for various prices of fuel and placed side by side with the figures of Table XV (p. 51), which is based on 100 per cent. efficiency. These are given in Part (*b*) of Table LXVII. It will be seen later that even these do not represent a perfectly true comparison except where the heating is continuous, and where the mains losses are equal in each case.

(*d*) **Period of Use**—Here we have to assess the occupancy of the building and the length of time the heating system must be at work. The cooling time of the building will have a bearing on this point; but it has been shown in Chapter III that for all ordinary buildings the cooling time of the building is so long that the wall-temperature will not drop appreciably overnight or even during a week-end. The necessity for reheating the air, before occupation, is allowed for by the preheating period mentioned below.

The usage may be resolved into periods per year, per week, and per day. For the yearly use a figure of thirty weeks is commonly assumed, i.e. from the end of September to mid-May.

The weekly use depends on the kind of building, whether seven days a week, 5½ (when week-ends are omitted) or only one or two days.

The daily use depends on the type of building. It will be appreciated that very few are heated continuously for twenty-four hours per day. The shorter the period of heating each day the greater will be the loss due to banking of the boilers if they are constantly alight, or the greater the preheating necessary in the morning if they are intermittently fired.

In other words, with coal or coke (hand stoked or magazine fed) the boilers will be banked a part of the day and all night with little result in the building. On opening up in the morning, however, less preheating is necessary after night banking, as the system will all be warm and the building will not have cooled off so much. The whole of the heat at night is not therefore wasted.

On the other hand, if the system is automatic stoker, oil firing, gas or electricity, no night running will be necessary (except perhaps in severe weather), but the preheating in the morning will be slightly longer. The assumption will be that the method of firing may be classed in two groups:

Continuous Firing, using solid fuel fed by hand or from magazines.

Intermittent Firing, using solid fuel with automatic stoking; or electricity, gas or oil fuel.

In addition, those buildings which are not occupied at week-ends will call for week-end banking, or for complete relighting early on Monday mornings, which will often take as much fuel as banking, with less desirable results. Whichever method is adopted there will be a longer period of preheating necessary on Monday mornings than on other days for these cases.

It is suggested that the 'period of use' may be divided into about six categories as follows:

(1) *Continuous Heating*. Hospitals, three-shift factories.

(2) *Continuous with Reduction at Night-Time.* Houses, flats, hotels, boarding schools.

(3) *Daytime Heating.* Offices, public buildings, shops, factories (one shift).

(4) *Part Daytime Heating*. Day schools.

(5) *Intermittent*—two days per week. Churches with mid-week meetings. Public halls used two days per week.

(6) *Intermittent*—one day per week. Churches, Sunday schools.

Many more intermediate divisions are no doubt possible, but these cover the majority of cases.

The allowance for preheating for the various categories, in terms of hours equivalent full use, may be taken as follows:

TABLE LXVIII

PREHEATING AND BANKING PERIODS

<table>
<tr><th rowspan="2">Type of Heat Requirement (See above)</th><th rowspan="2">Daily Preheating, Hours</th><th colspan="3">Continuous Firing</th><th rowspan="2">Intermittent Firing. Additional Preheating in Morning, Hours</th></tr>
<tr><th>Hours Banked at Night, at 15 Per Cent.</th><th>Extra Monday Morning Preheating, Hours</th><th>Hours Banked at Week-end, at 15 Per Cent.</th></tr>
<tr><td>1</td><td>—</td><td>—</td><td>—</td><td>—</td><td>—</td></tr>
<tr><td>2</td><td>1</td><td rowspan="5">24 hours, less occupation period, less preheating</td><td>—</td><td>—</td><td rowspan="5">1 hour per day of use</td></tr>
<tr><td>3</td><td>2</td><td>2</td><td rowspan="2">Total hours unoccupied, less preheating</td></tr>
<tr><td>4</td><td>3</td><td>2</td></tr>
<tr><td>5</td><td>6</td><td>—</td><td>—</td></tr>
<tr><td>6</td><td>8</td><td>—</td><td>—</td></tr>
</table>

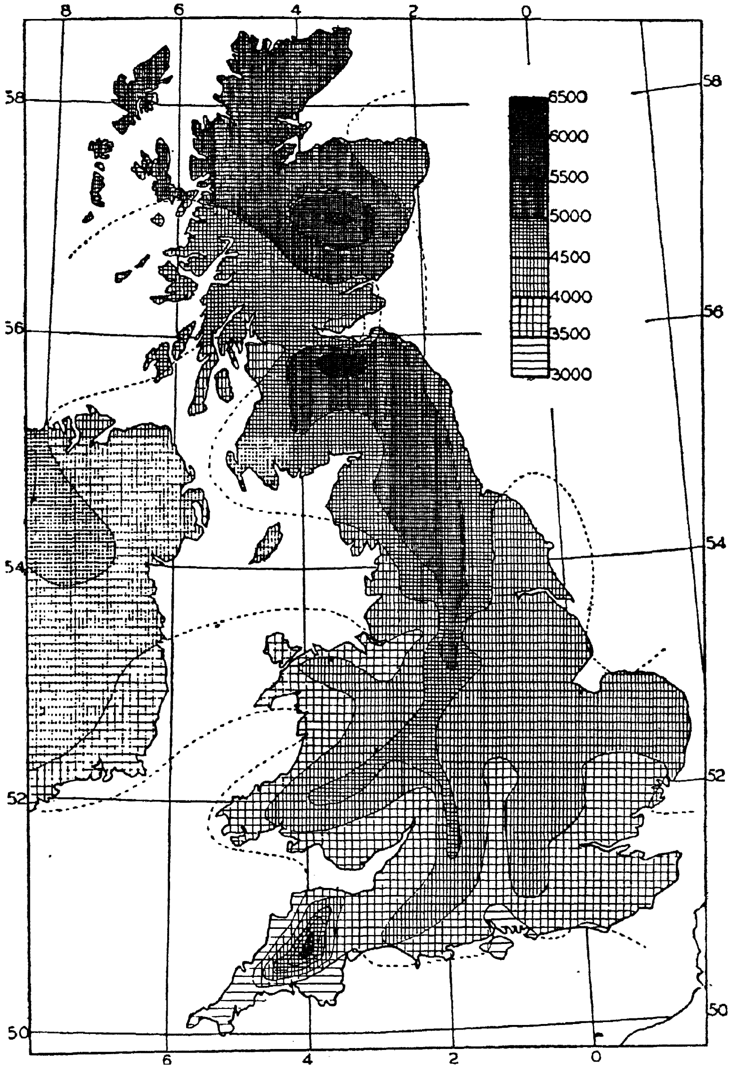


FIG. 219.—British Degree Days.

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(*e*) **Proportion of Full-Load Operation**—This will be a variable factor depending on the weather. Obviously no system will be called upon to operate at 100 per cent. output (based on 30° F. outside) during the whole season. What proportion of this full load can be assumed for the purpose of calculation, and how does it vary for different parts of the country?

Some interesting notes on this subject have been published* by Mr. A. F. Dufton, M.A., D.I.C., of the Building Research Station, and we reproduce here the map accompanying these notes (see Fig. 219).

The map shows 'British degree days'. The 'degree day' has proved a useful unit of reference in America, where it is the difference between 65° and the daily mean temperature when the latter is below 65°. Monthly normals of temperature (maximum, minimum and mean) are published by the Meteorological Office and the number taken is the mean annual total.

In America, as Mr. Dufton points out, 65° was chosen because this is the maximum at which heat is generally required (70° F. being inside temperature), and the amount of fuel burnt has been shown to follow almost directly the difference between 65° and the outside temperature.

In Britain an inside temperature of 65° F. is more common, and fuel is not usually consumed with the temperature over 60° F. This has, therefore, been taken as the base for the British degree day.

In his notes Mr. Dufton mentions that the effect of altitude will be seen in the case of the only two stations above 1000 ft. (Braemar 1120 ft. and Princetown 1359 ft.). Thus Dartmoor appears a cold spot, whereas Snowdon and Ben Nevis do not. However, mountainous districts are thinly populated, and no practical value would be achieved by attempting meticulous accuracy.

The degree day is not an absolute unit, since low temperatures at night produce more degree days than the proportion at that time may warrant. As a means of reference it is, however, a very useful measure.

Investigation of relationship between average temperatures and fuel consumption for a large number of Government Buildings has shown that:†

(1) There is a strong degree of correlation between fuel consumption and mean temperature.

(2) The degree of correlation is lower in mild weather.

(3) On the whole, the correlation with the mean 24-hour temperature is slightly stronger than that with the day temperature, whether for continuous or intermittent heating.

(4) The minimum temperature reached during a 24-hour period exerts a considerable influence on fuel consumption, particularly in mid-winter.

(5) For a given mean temperature, maximum temperature has a negligible effect.

(6) The effect of wind on fuel consumption rises to a maximum in the early part of the winter, and thereafter decreases steadily.

* See *Journal I.H.V.E.*, April 1934.
† A. C. Pallot, M.B.E., B.Sc., *I.H.V.E. Journal*, 1940–41, pp. 249–77.

(7) Fuel consumption, for the winter as a whole, tends to increase with an increase in the following factors:

Solar Radiation.
Barometric Pressure.
Rainfall.

(8) Fuel consumption tends to decrease with an increase in the following factors:

Atmospheric Humidity.
Cloud Amount.

Assuming that all the degree days are in the thirty weeks' heating season (this is not strictly correct since it is an annual computation), the total number of degree days possible, if the heating system has been designed for an outside temperature of 30°, will be $30 \times 7 \times (60-30) = 6300$ degree days.

Taking an average figure from the map of 3500 to 4000, say, 3750 for the London district, we find that the proportion of full load is:

$$\frac{3750}{6300} = 0{\cdot}6.$$

It is proposed in the calculations which follow to take a factor of 0·6 as the proportion of full load use or 'weather factor'. For other parts of the country this should be increased or reduced in direct proportion to the number of degree days.

TABLE LXIX

EQUIVALENT HOURS PER ANNUM FULL USE FOR VARIOUS HEAT REQUIREMENTS

Type of Heat Requirement					Total eek / Equi Hou B	uday Hour eating in		Total Week
1. Continuous - -		7	168					
2. Continuous except night-time - -	8 a.m.– 10 p.m.	7	105	9				
3. Daytime, excluding week-ends	9 a.m.– 6 p.m.	5½*	60½	13			40	
4. Part daytime - -	9 a.m.– 4 p.m.	5½*	55	14			39	
5. Intermittent, 2 days per week	10 a.m.– 8 p.m.		32	14†	4¼			
6. Intermittent, 1 day per week	10 a.m.– 8 p.m.		18	14‡	2			

* The half-day is presumed to be until 1 p.m.

† For use at 10 a.m. on Sunday, preheating would commence at 2 p.m. on Saturday and would go on until 6 p.m., followed by 14 hours banking, with a further 2 hours preheating before occupation commenced. This complete cycle occurs twice a week.

‡ A similar cycle to No. 5, but starting at noon on Saturday and occurring once a week.

TABLE LXIX—*continued*

Type of Heat Requirement		Continuous Firing						Intermittent Firing	
		Colu Fac			Number o Use per A				umn G Factor /eeks – E nt Hours per nnum
1. Continuous - -	168	0·6	101		101	3030	:68	168	3030
2. Continuous except night-time	105	0·6	63	9½		2175	05	112	2016
3. Daytime, excluding week-ends		0·6	37½|	15¾	53¼|	1597	62½	68½	1233
4. Part daytime - -	57	0·6	34	16½		1515	57	63	1134
5. Intermittent, 2 days per week	32	0·6	19	4¼		697	32	34	612
6. Intermittent, 1 day per week	18	0·6				390	18	19	342

Hours of Use per Annum—Table LXIX attempts to combine all the foregoing conclusions in a simple form.

The first part of the table shows the equivalent hours firing per week which would be required for useful heating (i.e. during the occupation period and for preheating) and for banking, assuming that a boiler can be kept alight at 15 per cent. of its normal consumption. These figures are all on the basis of '30-degree' weather conditions, i.e. weather factor = unity.

The second part of the table gives the hours firing per annum (30 weeks' heating season) with a mean weather factor of 0·6. It will be noted that this factor has not been applied to the figures for night banking, since this loss is constant at the 15 per cent. given, no matter what the weather may be.

It is admitted that there is, in a table like this, plenty of room for argument as to what allowances should properly be made, and it is put forward solely for what the authors believe to be average conditions met with in their experience. There can be nothing final or dogmatic about it, especially when divorced from any particular job in hand. It does, however, give a rational method for estimating the various factors, and, as will be seen later, appears to agree with the results obtained in practice.

Calculation of Fuel Consumption—The consumption is now simply a matter of arithmetic, and is given by the formula:

$$F \quad B \times E$$

where F = fuel consumption per annum in tons (coal, coke, oil), therms (gas), units (electricity).

B = maximum hourly B.T.U.'s output (converted to therms, 1 therm = 100,000 B.T.U.'s) of heating system including mains, calculated for 30° outside.

E = equivalent hours of full load use per annum (see Table LXIX).
T = therms per ton, therm, or unit, for fuel and method of firing selected (see Table LXVII).

The term E has been calculated on the basis of a weather factor of 0·6. If the degree days are more or less than 3750, a fresh factor must be calculated and applied as in Table LXIX to arrive at the correct number of equivalent hours.

Check Calculations—It is now proposed to calculate on the above basis the fuel consumption of five jobs of different types and check the result against the actual consumption of each.

Example 1

Hospital (Midlands)—continuous as type 1, 3030 hours.
Coal hand-fired—157 therms per ton.
B.T.U.'s per hour—2,145,000 = 21·45 therms.
(Cube 1,048,000 B.T.U.'s per cu. ft. = 2·0.)
Degree days 4000.

$$\text{Estimated fuel p.a.} = \frac{21{\cdot}45 \times 3030}{157} \times \frac{4000}{3750} = 437 \text{ tons.}$$

Actual fuel p.a. (1937-8) = 410 tons.

Example 2

Small Private House (Home Counties).
Continuous except at night as type 2, 2175 hours.
Anthracite hand-fired—170 therms per ton.
B.T.U.'s per hour 40,000 = 0·4 therms.
(Cube 25,000 B.T.U.'s per cu. ft. = 1·6.)
Degree days 3750.

$$\text{Estimated fuel p.a.} = \frac{0{\cdot}4 \times 2175}{170} = 5{\cdot}12 \text{ tons.}$$

Actual fuel p.a. (1937-8) = $5\frac{1}{2}$ tons.

Example 3

Office building (London).
Daytime heating as type 3, 1597 hours.
Furnace coke hand-fired—157 therms per ton.
B.T.U.'s per hour 1,800,000 = 18 therms.
(Cube 962,000 B.T.U.'s per cu. ft. = 1·9.)

$$\text{Estimated fuel p.a.} = \frac{18 \times 1597}{157} = 183 \text{ tons.}$$

Actual fuel p.a. (1937-8) = 162 tons.

Example 4

Bank, London.
Daytime heating as type 3 with intermittent firing, 1233 hours.
Oil fuel—302 therms per ton.
B.T.U.'s per hour 1,360,000 = 13·6 therms.
(Cube 915,000 B.T.U.'s per cu. ft. = 1·8.)

Estimated fuel p.a. = = 56 tons.

Actual fuel p.a. (1937-8) = 59 tons.

Example 5

Church and offices, London.
Intermittent two days a week for half the building as type 5, remainder as type 3—mean = 922 hours.
Oil fuel—302 therms per ton.
B.T.U.'s per hour 1,816,000 = 18·16 therms.
(Cube 933,000 B.T.U.'s per cu. ft. = 1·95.)

$$\text{Estimated fuel p.a.} = \frac{18{\cdot}16 \times 922}{302} = 55{\cdot}5 \text{ tons.}$$

Actual fuel p.a. = 60 tons.

It will be seen that so far as these buildings are concerned the calculated results agree well with the actual figures obtained.

Actual Fuel Costs—Table LXX gives fuel costs for heating and H.W.S. of a number of actual installations together with the cube and the cost per foot cube.

TABLE LXX

EXAMPLES OF FUEL COSTS PER ANNUM
(Heating Plus Hot-Water Supply)

Type of Building	Gross Cube (Cu. Ft.)	Fuel	Amount used per Annum	Cost per Ton or Unit	Total Cost per Annum	Annual Cost per Cu. Ft.
						d.
1. Banking Institution, London	5,000,000	Oil	520 tons	72/6	£1890	·091
2. Bank Head Office, London	3,129,000	Oil	269 tons	74/6	£1002	·080
3. Bank Head Office, London	2,000,000	Oil	183½ tons	80/-	£738	·089
4. Bank Head Office, London	2,000,000	Coke	218 tons	33/8	£367	·044
5. Bank Head Office, London	915,000	Oil	68 tons	86/-	£262	·068
6. Bank Head Office. London	2,250,000	Electricity (Thermal Storage)	1,500,000 units	0·3d. and 0·27d.	£1820	·194
7. Government Office Building, London	1,127,000	Oil	110 tons	72/6	£400	·085
8. Government Office Building, London	1,349,000	Oil	157 tons	75/-	£590	·105
9. Private Office Building, London	962,000	Coke	164 tons	32/6	£256	·067
10. Private Office Building, London	18,200	Anthracite	2¾ tons	55/-	£7 11s.	·10
11. Church, London	933,000	Oil	61 tons	86/-	£262	·068
12. Private House	120,000	Coal	12·35 tons	38/7	£23 16s.	·048
13. Private House	25,000	Anthracite	5·5 tons	55/-	£15 2s.	·145
14. Boarding School (South of England)	6,500,000	Coal	2050 tons	22/9	£2335	·087
15. Secondary School (West of England)	523,000	Coke and Anthracite	121 tons	29/3	£177	·081
16. Secondary School (Lancashire)	611,000	Anthracite	117½ tons	40/-	£235	·092
17. Hotel (London)	3,500,000	Coal	1500 tons	20/-	£1500	·10
18. Hotel (North of England)	425,000	Coke	260 tons	30/-	£390	·22
19. Hotel (Midlands)	6,200,000	Coal	2000 tons	20/-	£2000	·08
20. Hotel (Scotland)	3,250,000	Coal	1800 tons	20/-	£1800	·13
21. Infectious Diseases Hospital (Midlands) with Laundry, etc.	1,048,000	Coal	945 tons	16/7	£738*	·177
22. Block of 400 Flats (High Class)	4,250,000	Coke	1970 tons	31/6	£3100†	·175

* Cost per bed, £4 19s. 6d. per annum = 22·8d. per week/bed. L.C.C. average for nine fever hospitals = 32·4d. per week/bed. Ratio, heating cost : total cost (L.C.C.) 3·6 per cent.

† Cost per flat, £7·8 per annum.

(See note in Preface as to costs.)

Figures of this kind are not often published; indeed, it is uncommonly hard to come by them. They have been collected over a long period and are quite authentic.

It should be noted that the building cubes given are gross, i.e. including walls, roofs, basement, etc. That is why banks are relatively low per foot cube, since their large vaults increase the cube, but have practically no heating. Buildings of a type should be considered together, both for this reason and because their period of use is so different. Thus the electric thermal storage system in its own group will be seen to be over double the cost per foot cube of similar buildings heated by oil, whereas, comparing the figure with another group, such as a hospital with its continuous heating and absence of basements, etc., there might appear to be little to choose on cost.

These figures do not bring in the other factors mentioned on p. 390, i.e. labour, maintenance, etc., as the comparison would be too unwieldy.

Fuel Cost of Hot-Water Supply Systems—It is possible to calculate the fuel consumption for a hot-water supply system if the quantity of hot water to be heated per day or per week is known. The difficulty is that this is not known in advance, and different buildings appear to vary so vastly in this respect that any data collected on the point are of little use without interpretation. Typical figures of consumption were given on page 231, and may serve as a rough guide. The problem is least difficult where the demand is fairly steady, as in a large institution or hospital.

The fuel consumption is also dependent on the radiation from mains, towel airers, linen cupboard coils and other apparatus, and this continues twenty-four hours a day, all the year round, unless means are taken to shut off at night.

Taking a figure from Table XLV for an office building, we get 5 gallons per day per head for all purposes, maximum, or about 3 gallons average over the week. An example well known to the authors contains 300 occupants.

This gives $300 \times 3 = 900$ galls. per day.

[Raised 70° F. $\times$ 10 lbs. per gallon]	=	630,000 B.T.U.'s/day
Mains loss 50,000 B.T.U.'s per hour for 12 hours per day, dropping to, say, 50 per cent. mean at night $= 18 \times 50{,}000$	=	900,000 B.T.U.'s/day
		1,530,000 B.T.U.'s
		= 15·3 therms/day.
$5\frac{1}{2}$ days per week $\times$ 52 weeks		= 4400 therms p.a.

$$\text{The fuel consumption with furnace coke hand-fired} = \frac{4400 \text{ therms}}{157 \text{ therms per ton (Table LXVII)}}$$

$= 28$ tons per annum.

The actual consumption over several years has been 24 tons per annum, costing £38 10s. In this case the figures agree well, and the above example serves to show the method of calculation.

The building has a gross cube of 960,000 cu. ft., giving a cost of 0·01d. per cu. ft. per annum.

Comparative figures with other fuels could be estimated by using the appropriate heat value per ton from Table LXVII.

2. RUNNING OF AUXILIARIES

A further item in running costs is the electric current consumed by motors driving blowers, etc., for automatically fired boilers. This applies to oil firing, automatic coal stokers and to some types of magazine boilers and gas boilers. It does not occur in the case of natural draught boilers.

Motors for this purpose vary, but $\frac{1}{4}$ h.p. to $\frac{1}{2}$ h.p. generally serves boilers up to about 500,000 B.T.U.'s per hour, $\frac{1}{2}$ to 1 h.p. boilers up to 1,000,000 B.T.U.'s per hour and so on *pro rata*. Taking 1 kilowatt as roughly equal to 1 h.p. (including electrical losses), for a full heating period of 5040 hours the consumption for a motor of 1 h.p. = 5040 units per annum, at 1d. a unit for power = £21. With thermostatic control the motor may run perhaps no more than half the time, so that an expenditure of roughly £10 per annum for 1,000,000 B.T.U.'s per hour may be assumed.

Current is also required for the circulating pump, if any. For this about $\frac{1}{2}$ to $\frac{3}{4}$ h.p. would usually be required for 500,000 B.T.U.'s per hour and 1 h.p. for 1,000,000 B.T.U.'s per hour and so on. This, also, running continuously would cost £21 per annum for 1,000,000 B.T.U.'s per hour at the same price per unit. Where the pump is stopped at night or at week-ends this would be proportionately reduced. Thus for 12 hours a day an expenditure of £10 per annum per 1,000,000 B.T.U.'s per hour could be assumed.

It should be remembered that the heat equivalent of the power input to circulating pumps is not entirely wasted but comes out in the form of heat throughout the system. Of the power supplied as much as 60 per cent. may be delivered by the pump to the water as 'water horse power'. This is dissipated as heat in the friction of the pipes and is liberated for useful warming. Its proportion to the whole is, of course, so slight as to be unnoticeable.

In any case the cost of running of circulating pumps is common to all fuels and need not therefore appear in a comparative table.

3. LABOUR

Experience of a wide variety of systems appears to indicate the following labour requirements:

Hand Firing—(i) Small boilers, heating and hot-water supply—short periods three or four times a day: caretaker or odd labour only. Part time chargeable to heating, say, £25 per annum.

(ii) Up to three large boilers about 2,000,000 B.T.U.'s per hour each with daytime heating and night banking—one engineer who is required for general maintenance in any event, plus one whole-time stoker, say, £130 per annum. Part engineer's time might be charged to heating, costing, say, £50 per annum. Total = £180 per annum.

(iii) The same boilers as (ii) but in a block of flats or similar building where full heating to midnight is required, followed by night banking.

Daytime—part time of engineer's services only (say £50). Two shifts of stokers £260. Total £310.

(iv) The same as (ii) but full time heating as in hospitals, etc., one part-time engineer, three whole-time stokers extra. Total £440.

(v) Large boiler plants. One engineer and assistant plus one or two men for each of three shifts. The total for the eight men might amount to £1200-£1400 per annum. It has been shown that one man can stoke 1000 tons per annum.

Magazine-Fed Coke or Coal Boilers—Time taken is about ¼ to ½ hour per day per boiler. This means that up to four or five boilers (no matter whether heat is required all night or not) the duty is only a small part of the engineer's work. Again he cannot be dispensed with for other reasons, and £30 or £40 per annum should cover the proportion of his time devoted to heating. For a large number of boilers two men would probably be required part time, costing up to £150 per annum.

Automatic Coal Stokers—Except where hopper or direct feed is applied, approximately the same quantity of coal has to be shovelled as with hand firing. It is, however, done at about six-hour intervals instead of every three or four. Clinker and ash removal are the same.

The remarks applying to hand firing apply, therefore, to this system, except that the work is more of a part-time job.

Where hopper feed is used to all the stoker bins the case becomes similar to that of magazine boilers.

Oil Firing—(i) Small automatic plants are safely left unattended all day and night and labour is only required for cleaning the boiler flues once a week, cleaning the burner and general supervision. This is so slight as to be negligible.

(ii) Larger plants, other than power boilers, require part time cleaning and supervision by the normal engineer, for which, say, £50 per annum is chargeable. Night running can often be supervised by the night watchman if one is available. If not, an extra man is required for night-time, costing, say, £150 per annum.

(iii) Large plants consisting of several steam boilers—one man per shift must be in attendance in any event, and probably could not be dispensed with no matter what fuel was used. Cost, say, £400 per annum.

Gas Firing—It has already been stated that no labour is involved with this fuel, and no cleaning.

Supervision costs may be ignored, as they represent but a small fraction of caretaker's or engineer's time.

Electric Thermal Storage—The same remarks apply as with gas.

4. MAINTENANCE AND REPAIRS

For comparative purposes, this item need only be taken as referring to the boiler plant. The ordinary pipes and radiators call for no maintenance and no repairs, except occasional packing of valves or on account of trouble due to frost.

The maintenance and repairs necessary for the boiler plant are difficult to give in figures, but might be summarized as follows:

Hand Firing—Replacement of grate bars once every three years. If the bars are water-cooled, this does not arise. Cost might be £5 per annum for a large boiler, including servicing of automatic controls.

Magazine Boilers—These are the same as for hand firing in this respect.

Automatic Stokers—The fire-pot and worm may call for renewal every three or four years. The gearing will require periodical overhaul. Automatic controls should be serviced once or twice a year. Probably £10 per stoker per annum should be allowed, but this depends on the make, quality of coal and method of running.

Oil Firing—Overhaul two or three times a year is desirable with self-contained units, and periodical renewal of electrodes, burner nozzles, brushes, etc., is necessary. All-in cost, say, £5 per burner per annum. Larger plants are generally attended to by the engineer, and the cost would be, if anything, less than above.

Gas Firing—Practically no maintenance is called for here, and its cost is negligible.

Electric Thermal Storage Systems—Renewal of electrodes once every three or four years may be necessary in the case of electrode boilers, and renewal of elements in the case of immersion heaters may be required at the same intervals. As the electric controls are complicated, their annual overhaul is essential. Average cost with any type, say, up to £20 per annum per boiler.

Generally it will be seen that none of the above represents a high proportion of total running cost. A figure sometimes taken is 5 per cent. of the installation cost, though this would appear to be too high. Repair or replacement of the boiler is not included, as this is really common to all systems in some form or another.

5. INTEREST AND DEPRECIATION

This, again, when comparing fuel costs, should not apply to the heating system proper, but only to the additional cost of the equipment necessary for the particular fuel or method of handling.

If any difference is made in builders' costs by the adoption of an alternative fuel, interest on this also should be included.

These charges may generally be based on interest 5 per cent., depreciation 5 per cent. Items such as coal stokers, which may require almost complete renewal in ten years or so, should be debited with 10 per cent. depreciation.

Various costs of magazine boilers, stokers, oil-firing plants, etc., have been given, but in order to link them up with some concrete cases, the costs of four actual jobs have been taken out in Table LXXI.

TABLE LXXI

FIRST COST OF HEATING SYSTEMS

Type of Building	Gross Cube (Cu. Ft.)	Cost of System using Coal or Coke, hand-fired, in Cast-Iron Boilers, with:				Extra Cost (over Cast-Iron Boilers, hand-fired) for:				
		Hot-Water Radiators	Rayrads	Concealed Panels, Iron Pipes*	Concealed Panels, Copper Pipes*	Gravity Feed (Coke) Magazine-Boilers, including Steel Hoppers	Automatic Coal Stokers, hand-fired	Oil-firing, including Storage Tanks	Gas Boiler	Electrical Thermal Storage
		£	£	£	£	£	£	£	£	£
A. Small Hotel	165,000	560	740	700	840	275	190	340	160	500
B. Secondary School -	500,000	1900	2600	2580	3280	380	420	500	200	900
C. Municipal Offices -	1,270,000	2800	4000	3700	4200	550	470	650	250	1800
D. Hospital -	2,000,000	6500 (W.I. Boiler)	8700	8500	9900	750	660	1100	400	2300

* Does not include extra cost of special plastering. (See note in Preface as to costs).

The first price column relates to the cost of the heating system, including plain hand-fired boilers, pipes and radiators. Alternative prices are then given for the same building heated by Rayrads, and panels with iron or copper pipes. For comparison of fuel costs, no interest or depreciation is taken on these figures but on the ones following—i.e. on the extra sums due to magazine boilers, coal stokers, oil-firing, gas or electric boilers. No attempt has been made to estimate differences in cost of builders' work with the alternative schemes, as it varies so much according to conditions. For present purposes the builders' work for each is assumed the same.

6. INSURANCE

Insurance of hot-water or low-pressure steam heating boilers is not compulsory, as it is for high-pressure steam boilers.

Insurance of heating and hot-water supply boilers does, however, mean that any act on the part of the attendant due to ignorance or neglect, such as the overloading of a safety valve, is not allowed to go unchecked. A sense of security to the owner is given by the annual inspection of the

insurance company's engineer, and he knows that if anything is wrong, or if the boiler wants cleaning out, he will be told so.

A further point is that, should he be unlucky enough to be the one man in a million who does have an explosion, due, perhaps, to a stoker's folly, and the man is fatally injured, no reflection can come on him in any subsequent enquiry if the boiler had been regularly inspected.

One company quotes 50s. per annum for a cover of £5000 against explosion, third-party risk and damage to surrounding property, in respect of a single boiler of mild-steel construction. The rates for cast-iron boilers are somewhat higher.

These costs are, however, trifling, and may be ignored in any comparison of fuel costs, since they apply to all types in some form.

EXAMPLES OF RUNNING COSTS

It is impossible to devise a simple comprehensive formula embodying all the above factors, and the only practicable comparisons are between specific cases.

In order that the question of running costs may be brought to a conclusion, the annual expenses with different fuels for four buildings have been estimated and are given in Table LXXII (p. 407-8), these being the same examples as those referred to in Table LXXI. The four cases are of different character, with various methods of firing and times of use.

It should be noted that the calculated figures are for heating only, and that they allow for mains losses, the latter being included in the total heat required per hour.

The results bear out the previous conclusion, that with continuous heating the cheaper fuels are those which are cheapest per unit of heat, but with more intermittent use, such as in the school example, the differences with various fuels are much less marked.

COMPARISON WITH DIRECT SYSTEMS

So far, only systems of the indirect category have been considered, i.e. those with a boiler, and pipes for the conveyance of the heat to the rooms, and the figures in Table LXXII are, strictly, comparative boilerhouse running costs.

As has been pointed out with gas and electricity, Chapters XII and XIII, direct heating starts with the advantage of having no boiler losses, and no wastage of heat from mains. Further, it has no night banking loss, and is instantly responsive to thermostatic control. If a room rapidly fills with people, or the sun comes out, it will be found that the thermostat has shut off the heating. With a hot-water system overheating results, as even if individual room control is provided (which it rarely is) the time lag of the system is too great to take advantage of these rapid changes. This is particularly the case with schools, where the introduction of twenty or

thirty children to a classroom is often enough to keep it warm in mild weather without artificial heat.

TABLE LXXII

RUNNING COSTS OF FOUR TYPICAL BUILDINGS HEATED IN VARIOUS WAYS

(i) Fuel Costs

Building and Type of Heat Requirement (Cf. Table LXXI)	Heat required in B.T.U.'s per Hour	Hours per Annum Continuous Firing (Table LXIX)	∴ Therms per Annum Continuous Firing (Systems *a* and *b*)	Hours per Annum Intermittent Firing (Table LXIX)	∴ Therms per Annum Intermittent Firing (Systems *c*, *d*, *e* and *f*)
A. Small Hotel: Type 2, Continuous except night-time	393,000	2175	8,550	2016	7,920
B. Secondary School: Type 4, Part daytime	967,000	1515	14,640	1134	10,970
C. Municipal Offices: Type 3, Daytime, excluding week-ends	1,740,000	1597	27,800	1233	21,460
D. Hospital: Type 1, Continuous	3,000,000	3030	90,900	3030	90,900

Annual Fuel Consumption at Rates given in Table LXVII, also Cost* for Various Types of Heating

Building and Type of Heat Requirement (Cf. Table LXXI)	(*a*) Coke, Hand-fired at 157 Therms/Ton	(*b*) Coke in Magazine Boiler at 182 Therms/Ton	(*c*) Coal with Automatic Stoker at 184 Therms/Ton	(*d*) Oil at 302 Therms/Ton	(*e*) Gas Boiler at 0·85 Therms/Ton	(*f*) Electrical Thermal Storage at 0·0335 Therms/Unit
A. Small Hotel: Type 2, Continuous except night-time	54 tons £94	47 tons £82	43 tons £54	26 tons £111	9,300 therms £155	236,000 units £246
B. Secondary School: Type 4, Part daytime	93 tons £163	81 tons £141	60 tons £75	36 tons £153	1,290 therms £215	328,000 units £342
C. Municipal Offices: Type 3, Daytime, excluding week-ends	177 tons £310	153 tons £268	117 tons £146	71 tons £302	25,300 therms £422	642,000 units £670
D. Hospital: Type 1, Continuous	580 tons £870	500 tons £750	493 tons £618	300 tons £1162	107,000 therms £1,780	2,700,000 units £2,800

* Based on following fuel costs: coke 35s. per ton (30s. for large quantities), coal (for automatic stoker) 25s. per ton, oil 85s. per ton (77s. 6d. over 100 tons per annum), gas 4d. per therm, electricity ¼d. per unit. (See note in Preface as to costs.)

TABLE LXXII—*continued*

(ii) Inclusive Costs, per Annum

Building and Type of Heat Requirement	Type of Heating	Fuel Cost (from (i) above)	Electric Power for Firing	Labour	Maintenance	Interest and Depreciation (on extra cost over C.I. Hand-fired Boilers)*	Total Annual Cost
A. Small Hotel: Continuous, except night-time	(*a*) Coke, Hand-fired	£94	—	£25	£5	—	**£124**
	(*b*) Coke in Magazine Boiler	£82	—	£12	£5	£28	**£127**
	(*c*) Coal with Automatic Stoker	£54	£4	£12	£10	£29	**£109**
	(*d*) Oil	£111	£4	—	£5	£34	**£154**
	(*e*) Gas Boilers	£155	—	—	£2	£16	**£173**
	(*f*) Electrical Thermal Storage	£246	—	—	£10	£50	**£306**
B. Secondary School: Part daytime	(*a*) Coke, Hand-fired	£163	—	£80	£10	—	**£253**
	(*b*) Coke in Magazine Boiler	£141	—	£30	£10	£38	**£219**
	(*c*) Coal with Automatic Stoker	£75	£10	£50	£20	£63	**£218**
	(*d*) Oil	£153	£10	£30	£10	£50	**£253**
	(*e*) Gas Boilers	£215	—	—	£5	£20	**£240**
	(*f*) Electrical Thermal Storage	£342	—	—	£20	£90	**£452**
C. Municipal Offices: Daytime, excluding week-ends	(*a*) Coke, Hand-fired	£310	—	£130	£15	—	**£455**
	(*b*) Coke in Magazine Boiler	£268	—	£40	£15	£55	**£378**
	(*c*) Coal with Automatic Stoker	£146	£20	£80	£30	£70	**£346**
	(*d*) Oil	£302	£20	£50	£15	£65	**£452**
	(*e*) Gas Boilers	£422	—	—	£10	£25	**£457**
	(*f*) Electrical Thermal Storage	£670	—	—	£30	£180	**£880**
D. Hospital: Continuous	(*a*) Coke, Hand-fired	£870	—	£440	£30	—	**£1340**
	(*b*) Coke in Magazine Boiler	£750	—	£150	£30	£75	**£1005**
	(*c*) Coal with Automatic Stoker	£618	£50	£200	£40	£100	**£1008**
	(*d*) Oil	£1162	£50	£200	£20	£110	**£1542**
	(*e*) Gas Boilers	£1780	—	£50	£15	£40	**£1885**
	(*f*) Electrical Thermal Storage	£2800	—	£50	£40	£230	**£3120**

* Based on First Costs from Table LXXI. Interest, 5 per cent. all cases; depreciation, 5 per cent. all cases, except (*c*), which is ten per cent.

(See note in Preface as to costs.)

Direct Electric Heating—Electricity is eminently suited to thermostatic control to meet rapidly changing conditions, and it is quite reasonable to anticipate a much lower heat input to suffice in such cases, with electricity than with any other fuel.

Some figures stated to be taken from a South of England Education Committee's report appear to confirm this. Two schools for 480 children, one heated by electricity direct and one by oil, consumed in the year:

Electricity	Oil
54,200 units, equivalent to 1860 therms. Cost at 0·75d. =£150.	23 tons, equivalent to 6946 therms. Cost at 85s. per ton =£98.

This gives roughly four times the heat input with oil as compared with electricity, yet results are stated to be equally satisfactory.

As regards first cost, both were priced at £800, but the hot-water scheme is believed to have cost £700 more for builders' work.

Further running consumptions for a number of schools per cubic foot (gross) have been submitted to the authors as follows:

	Units per Cu. Ft. per Annum
Schools, London	0·31
,, ,,	0·29
,, Belfast	0·276
,, Cambridge	0·24
,, Torquay	0·26
,, Glasgow	0·204
,, Kent	0·294

From the above a figure of 0·3 might be assumed to be normal. This consumption at 0·75d. a unit gives a cost of 0·25d. per ft. cube, compared with the two coke-fired schools given in Table LXX at 0·09d. approximately.

Taking labour, maintenance and interest on builders' work into account, it may be shown in some cases that a fairly close comparison is possible, but in the one case (hot water) some overheating probably results, in the other (electricity) it is precise and limited.

This comparison for schools is as favourable to electricity as any, for reasons already stated. It will be clear that the more nearly continuous the heating, the less favourable will be the case for electricity.

Some figures obtained for a private house are given below. It is heated by electric radiant ceiling panels to 55° for living rooms and 45° for bedrooms, augmented by direct electric radiators.

Water heating is by electric storage heater.

The radiant panels are thermostatically controlled, and it is found that a saving results by shutting off at night.

Gross cube, 20,000 cu. ft.

		Unit per Cu. Ft. Gross
Average for 2 years:		
Heating, lighting, cooking	11,790 units	=0·59
Water heating	7,320 ,,	=0·37
	Total	0·96

The current is obtained at ½d. a unit in summer, and ¾d. a unit in winter for all purposes, plus a standing charge of £4 18s. per annum. Water heating is priced at ½d. a unit all the year, and is cut off by a time switch at peak load.

After deducting cooking and lighting, the annual cost for heating and hot water is approximately £45 per annum.

This gives a cost of 0·54d. per foot of gross cube, which will be seen to be over three times the cost given in Table LXX of a similar sized house heated by a combined heating and hot-water system fired with anthracite.

Direct Gas Heating—Owing to the wide difference in type between the various forms of gas heating, any comparison with running costs of indirect systems is meaningless without specifying exactly the direct gas system to be employed. The basis of calculation for the specific cases must of course be the efficiency to be assumed. This matter has been referred to more fully in Chapter XII on Gas Heating.

Comparison of Running Costs—The running costs of direct electric and gas heating for the four examples given in Table LXXII may be calculated for comparison with the indirect systems by taking the hourly heat input without mains losses, and allowing 100 per cent. and 60 per cent. efficiency respectively. The time of use may be taken the same as before for intermittent firing, with a deduction of 25 per cent. for thermostatic room control.

It is assumed in both cases that there are no labour, maintenance or other standing charges. On this basis the annual costs are arrived at thus:

TABLE LXXIII

RUNNING COSTS WITH DIRECT HEATING

Building	Net Heat Requirements in B.T.U.'s per Hour	Hours per Annum (as Intermittent Firing, Table LXIX)	25% Deducted for Thermostatic Room Control					
			Electricity at 100% Efficiency			Gas at 60% Efficiency		
			Units per Annum	Cost at ½d./unit	Cost at ¼d./unit	Therms per Annum	Cost at 8d./therm	Cost at 4d./therm
A. Small Hotel	328,000	2016	148,000	£308	£154	8,250	£275	£137
B. Secondary School	806,000	1134	187,000	£390	£195	10,600	£353	£176
C. Offices	1,450,000	1233	393,000	£820	£410	22,400	£747	£373
D. Hospital	2,500,000	3030	1,660,000	£3460	£1730	94,700	£3160	£1580

(See note in Preface as to costs.)

CHAPTER XVII

Ventilation

The word *Ventilation* means literally the causing of air movement or wind (L. *Ventus*), but has acquired the meaning of a system which gives a regulated supply of air to an enclosed space so as to make the conditions better for human habitation.

The purpose of ventilation is fundamentally to supply the untainted air necessary for human existence, because life depends on a constant supply of oxygen. The supply of this air involves the removal of a corresponding volume of expired or vitiated air: with the smells and noxious gases which are associated with concentrations of people.

The normal adult at rest inhales about 1·8 cu. ft. of air per hour. Of this about 5 per cent. is absorbed as oxygen by the lungs. The exhaled breath gains from 3 to 4 per cent. of carbon dioxide (CO_2), equal to about 0·6 cu. ft. per hour. Thus, if it is desired to keep the concentration of CO_2 down to, say, 10 parts per 10,000, with external air containing 4 parts per 10,000, it will be necessary to supply:

$$\frac{\cdot 6}{\cdot 001 - \cdot 0004} = 1000 \text{ cu. ft. air per hour per occupant.}$$

In addition, the human body at normal temperatures at rest gives out about ·11 lbs. of water vapour and 300* B.T.U.'s per hour of sensible heat. The 1000 cu. ft. per occupant as above will become warmed and humidified as follows:

$$\text{Temperature Rise} = \frac{300}{1000 \times \cdot 019} = 15{\cdot}8^\circ \text{ (assuming no heat loss from room).}$$

$$\text{Increase in Humidity} = \frac{\cdot 11 \times 7000}{1000} = \cdot 77 \text{ grains moisture per cu. ft.}$$

It has long been accepted that the CO_2 content is not a satisfactory criterion of good ventilation. The CO_2 content may be as much as 20 or 30 parts per 10,000 and yet the air feel fresh and pleasant, or it may be under 10 and yet feel stuffy owing to other more important effects.

To state what is a satisfactory criterion is more difficult. Freshness of air in a room may be judged by the sense of smell, an analytical method more delicate than any laboratory one. Freshness or stagnation are in addition judged by the rate of loss of heat and moisture from the skin. This aspect of comfort has been referred to earlier under the discussion on 'Effective Temperature'.

* This varies greatly with Temperature. See Fig. 249.

The interplay of air temperature, air movement, and radiation have been referred to under Equivalent Temperature (p. 39). Where heating to a room is from a radiant source, the air temperature requires to be lower and the air movement greater than in a room with the same equivalent temperature, but with air and surfaces all at a uniform temperature, for the same degree of comfort.

The inter-relation of all these factors, as depicted in Fig. 220, is very complicated, and nothing short of an exhaustive investigation of each one

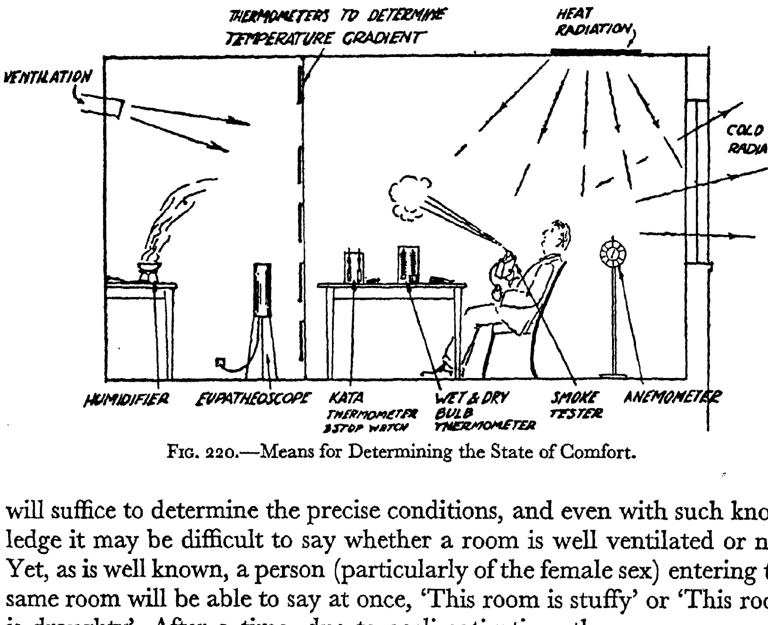


FIG. 220.—Means for Determining the State of Comfort.

will suffice to determine the precise conditions, and even with such knowledge it may be difficult to say whether a room is well ventilated or not. Yet, as is well known, a person (particularly of the female sex) entering the same room will be able to say at once, 'This room is stuffy' or 'This room is draughty'. After a time, due to acclimatization, the same room may appear to have become more tolerable.

The practice of ventilation is made further somewhat indefinite by the lack of an easy method of detecting slight air currents. A speed of about 2 ft. per second is generally regarded as the maximum which can be allowed without a feeling of draught. Such relatively small velocities can be measured by the Kata Thermometer (p. 40) or by the Hot Wire Anemometer, in which air flow is measured by means of the cooling effect on a heated wire. These do not, however, indicate direction, for which some visual detector is necessary, such as a cold smoke produced by the mixture of ammonia and hydrochloric acid fumes. Smoke, however, tends to diffuse rapidly and is useful only as a local index.

One velocity may be permissible if the direction is towards the face, but may constitute a draught if impinging on the back of the neck.

Good ventilation cannot, therefore, be defined in simple terms, and

reference has to be made to conditions which have been found in practice to give reasonably satisfactory results. These may be discussed under the following headings:

Volume of air necessary.
Distribution.
Temperature.
Humidity.
Purity.

VOLUME OF AIR NECESSARY

(*a*) **Temperature Rise Basis**—Where the occupancy is known the temperature rise can be estimated as with Air-Conditioning (see Chapter XVIII). If a rise of, say, 15° is accepted as reasonable and 300 B.T.U.'s per hour are assumed emitted per occupant, the volume of air required will be

$$\frac{300}{15 \times \cdot 019} = 1050 \text{ cu. ft. per hour per occupant.}$$

This assumes no building heat losses or gains other than from occupants.

It frequently happens that in crowded rooms much heat is also released from electric or other lamps, motors, stoves, cooking appliances, plant, etc.

The heat from these can generally be calculated, and a further air-change is needed to carry this extra heat away (see pp. 42-3).

In such cases the air per person per hour may greatly exceed 1000 cu. ft. per hour.

For summer a higher rate may be necessary, as there may be heat gains from the sun and a higher inlet air temperature, but it will be seen from a consideration of Chapter XVIII that the removal of this heat gain without cooled air calls for very large ventilation volumes, which may be undesirable from other points of view and unnecessarily high for winter.

(*b*) **Legislation Basis**—For theatres and music halls, including all places licensed for music and dancing, boxing, etc., a minimum of 1000 cu. ft. of fresh air per hour per occupant is required by the London County Council. Other similar bodies have adopted the same standard.

Factories where manual labour is employed for gain are governed by the Factories Act 1937. This does not, however, lay down numerical standards. H.M. Factories Inspectors interpret these requirements according to each separate case, but it may be said that normally six changes per hour are considered adequate. Exceptions occur where much gas is burnt and where cellulose vapours or noxious fumes occur. In aeroplane dope shops thirty changes per hour are called for. In the case of textile factories special requirements cover the humidity permissible. Reference may be made to

TABLE LXXIV

VENTILATION ON AIRCHANGE BASIS

Type of Room	Airchanges per Hour
Offices above ground	2–6
Offices below ground	10–20
Factories, large open type	1–4
Factories and workrooms closely occupied	6–8
Workshops with unhealthy fumes	20–30
Laundries, dye-houses, spinning mills	10–20
Kitchens above ground	10–20
Kitchens below ground	20–60
Lavatories	6–12
Boilerhouses and enginerooms	10–15
Foundries, with exhaust plant; rolling mills	8–10
Foundries, without separate exhaust plant	10–20
Laboratories	6–12
Hospital operating rooms	10
Hospital treatment rooms	6
Restaurants	8–12
Smoking rooms	10–15
Stores, strong-rooms	1–2
Assembly halls	3–6
School class rooms	3–4
Living rooms	1–2
Sleeping rooms	1
Entrance halls and corridors	3–4
Libraries	2–4

Welfare Pamphlet No. 5, *Ventilation of Factories and Workshops*, published by H.M. Stationery Office.

(*c*) **Airchange Basis**—Where the occupancy is unknown or variable, an arbitrary basis must be taken. The same must be the case where fumes, vapours, smoke, steam, etc., arise from the work carried on in the space. The above table may be used as a guide.

(*d*) **Effect of Cubic Content on Ventilation Rate**—The cubic content per occupant and length of time occupied obviously have an important effect on the rate of air supply necessary. Thus, in a large hall of 500,000 cu. ft. capacity, seating 500 persons, each will have 1000 cu. ft. of air already stored in the building. If occupied for three hours and the desired rate of supply is 1000 cu. ft. per hour per occupant, the actual fresh air needed is only

$$500 \times 1000 - \frac{500{,}000}{3} = 333{,}333 \text{ cu. ft. per hr.}$$

$$= 666 \text{ cu. ft. per hour per occupant instead of } 1000.$$

(*e*) **Odours**—One of the essentials of good ventilation is the removal of odours arising from human occupation. The problem only becomes serious in crowded places. A supply of fresh air at the rate of 600 cu. ft. per hour per person is found to be the minimum to obviate trouble from this

source. With workpeople in their dirty working-clothes, the figure may need to be three times as great.

DISTRIBUTION

The admission of fresh air to a room should be such that:

(*a*) It is evenly diffused over the whole area at the breathing level.

(*b*) It should not strike directly on the occupants.

(*c*) It should avoid stagnant pockets.

Various methods of approaching this end are discussed in Chapter XIX.

The extraction of vitiated air from the room has little directional effect apart from a general upward, downward or crosswise motion.

Distribution often determines the air volume to be circulated. Though a small quantity of fresh air may suffice in the case of a large room sparsely populated, it may be impossible to diffuse this evenly throughout. The volume must then be increased, either all as fresh air, or by re-circulating room air mixed with the small quantity of fresh air.

Similarly, in a small crowded room, distribution difficulties may prevent admission of the necessary airchange rate without draught, and the volume then has to be reduced with unavoidable rise of temperature unless full air-conditioning with cooling is employed.

Distribution limits appear to be about 3 airchanges per hour minimum, and 20 airchanges per hour maximum where mechanical means are employed. With natural inlet, lower rates than 3 will, of course, be possible and often sufficient, but distribution can then be only a matter of chance.

TEMPERATURE

The temperature of the air admitted must be not much below that of the room, or the air will fall to the floor without proper mixing and cause cold draughts. The air must not on the other hand be too warm, or it will rise to the ceiling without adequate distribution.

In a 'straight' ventilation system, which is here under discussion, it is assumed that the warming of the building is accomplished by a separate radiator or other direct heating system to supply the heat necessary to balance the heat losses in winter, say to 60 or 65°. Under these conditions the air supply will preferably be about 5° lower than the desired room temperature and, when occupied, the leaving air will be perhaps 10° or so higher.

Under summer conditions without provision for air-cooling, no control can be exercised over the inlet temperature except by the small reduction possible by humidification referred to later.

HUMIDITY

The fresh ventilation air admitted from outside will have the same moisture content as outside. In cold weather this may result, as has been shown earlier, in an unduly low relative humidity after heating. This may be adjusted, where mechanical inlet is provided, by humidification by means of water sprays or other device for adding water vapour, and the amount of moisture added may be controlled to give the desirable relative humidity of about 55 to 65 per cent. at room temperature.

In normal mild weather the outside humidity will generally be satisfactory without alteration. The human body is not critical of humidity variations at normal temperatures.

There is, further, a reservoir effect caused by the hygroscopic contents of buildings, such as fabric, wood, plaster, etc., which steadies up humidity changes, tending to equalize them over long periods.

In summer, when the outside humidity is high, there can be no control of the inside humidity without full air-conditioning which includes means for de-humidification. An air washer or other humidifying device can only *add* moisture, and due to evaporation may cool the air to a temperature approaching the wet-bulb.

Thus, with outside air at, say, 75° dry bulb and 65° wet bulb (relative humidity 60 per cent.), a single bank air washer may be able to reduce the temperature to 67°, at the same time increasing the relative humidity to 90 per cent. This final condition may be more oppressive than the initial, and it is indeed often found that in hot weather, air washers, where provided, are shut off for this reason. Reference to Fig. 15 (p. 40) will show that the above initial condition is just inside the summer comfort zone, but the latter condition is well outside it.

There are, however, systems designed specially for the purpose of cooling air by evaporation. These generally use the capillary cell type washer (see Fig. 336), capable of giving high saturation to the air, coupled with an abnormally high rate of airchange.

It is found that with high humidities much greater air-speeds are permissible in a room without sensation of draught, so that room heat gains can be removed by large volumes of air in this condition without recourse to artificial cooling. In the height of a heat wave with high wet bulb outside, such a system is powerless to effect appreciable cooling and such must be recognized as one of its limitations. It has been much used in America, but in this country where high humidities are common it is of doubtful application.

AIR PURITY

Most buildings requiring special ventilation are in the centre of towns where atmospheric pollution is highest. Such pollution is produced chiefly by the burning of coal and other fuels, and consists of soot, tar, ash and sulphur dioxide. The total solids deposited per month given by the Atmo-

spheric Research Pollution Committee averaged for the three years 1936-1939 the following rates:

	Total Solids per Sq. Mile per Month
London (City) - - - -	22·8 tons
Kew - - - - -	10·2 „
Glasgow (Centre) - - -	
Stoke-on-Trent - - - -	
Towns in country districts (various)	4 to 10

When air is introduced into a building for ventilation a proportion of these solids will be brought in also. If the inlet is natural, nothing satisfactory can be done to prevent this, but where mechanical inlet is provided, filters and washers may be employed to remove the greater part. The various devices available are discussed in Chapter XIX.

The sulphur dioxide cannot be arrested by any commercial system yet invented. Large power stations are now in some cases equipped with plant for removing these gases from the flue outlets of the boilers, but such is not applicable to ventilation.

METHODS OF VENTILATION

Ventilation may be:

Natural, in which air movement is induced by effect of temperature difference or wind.

Mechanical, in which air movement is caused by power-driven fan or fans.

Inlet and extract have to be considered separately.

There are thus four possibilities.

	Inlet	Extract
(1)	Natural.	Natural.
(2)	Natural.	Mechanical.
(3)	Mechanical.	Natural.
(4)	Mechanical.	Mechanical.

(1) **Natural Inlet and Extract**—This applies to most rooms and buildings with low occupancy, such as living rooms, bedrooms, small offices, schools, hospitals, shops, etc., and even then is dependent on clean outside air and conditions which permit of ample open windows.

If an open fireplace or gas fire with flue is provided, the flue serves the double purpose of carrying away the products of combustion and of exhausting air from the room (Fig. 221). This air is replaced by cold air drawn through cracks around doors and windows, between floor-boards, etc., thus causing ventilation of the room, though often at an unnecessarily high rate causing draughts.

Where no flue exists, as with a central heating system or electrical heating, natural ventilation is afforded by opening of windows, by fresh-air

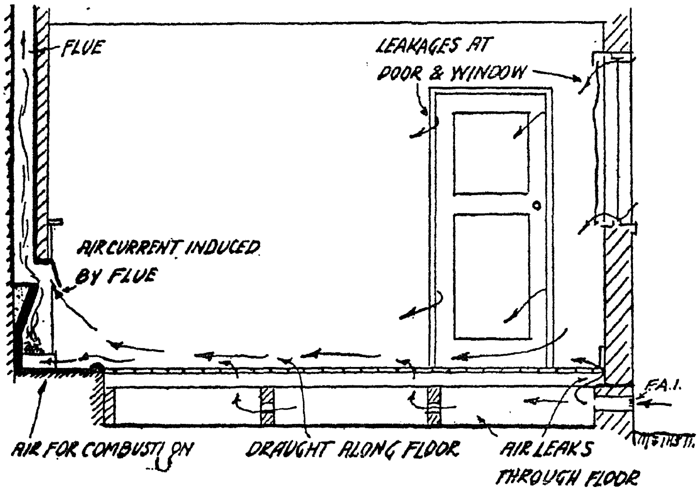


FIG. 221.—Ventilation by Open Fire.

inlets behind radiators or by wall gratings. Each of these methods has its objections, and there has yet to be devised an entirely satisfactory method. Thus, more often than not, no special provision is made for ventilation and natural infiltration alone causes what change of air there is, leading to an increasing deadness and staleness of the atmosphere as the day draws on.

A useful idea where windows are of sash type is the fitting of a raised sill, as shown in Fig. 222, permitting air to enter but destroying its velocity.

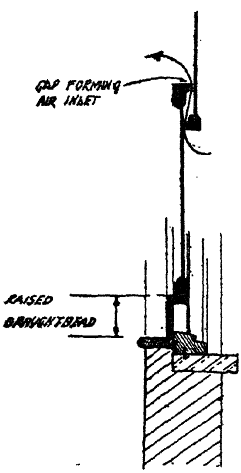


FIG. 222.—Ventilation with Sash Window.

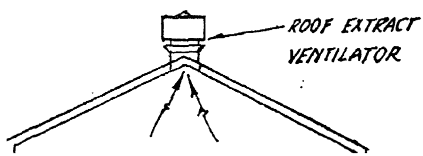


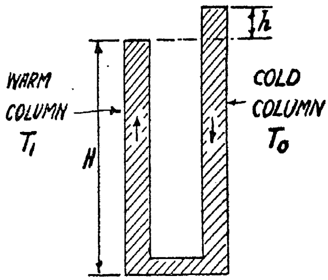
FIG. 223.—Ventilation by Natural Inlet and Extract.

For larger rooms, assembly halls, factories, etc., roof ventilators combined with fresh-air inlets behind radiators (Fig. 223) provide a cheap solution, but the operation is spasmodic and unreliable, depending partly

on temperature difference and partly on wind. Thus, in hot still weather when ventilation is perhaps needed most, no appreciable air flow occurs. In freezing weather with high winds the fresh-air inlets cause draughts, are frequently found permanently closed.

Consider the warm air inside a flue with cold air outside as forming two legs of a U-tube.

Outside temperature (absolute) T_0
Inside ,, ,, T_1



The density of air at the two temperatures is proportional to the absolute temperatures.

If the height of column T_1 is H, then the cold column, being denser, will in terms of the density of T_1 be equivalent to H plus some increased height h such that

$$h = H \; T_1 - T_0 .$$

This difference between the height of the two columns causing flow is

h in the usual formula $v = \sqrt{2gh}$,

where v = velocity and

g = acceleration due to gravity.

Then $v = \sqrt{2 \cdot g \cdot H}$.

This applies to any flue in which it is assumed atmospheric pressure is exerted at the top. In the case of a building with inlet at floor and outlet at ceiling, it is generally considered that a neutral zone at atmospheric pressure exists about halfway between floor and ceiling; the lower half is at an increasingly negative pressure from the neutral zone downwards, thus causing inward flow, and the upper half at an increasingly positive pressure from the neutral zone upwards to the ceiling, causing outward flow. The top of the imaginary U-tube must therefore in this case be at the zone of atmospheric pressure, which is at a level of $\frac{H}{2}$.

Then $$\frac{H}{2} \cdot \frac{T_1 - T_0}{T_1}$$

If v is in ft. per second, $g = 32 \cdot 2$ ft. per sec. per sec., H is in ft.

Table LXXV, calculated for an outside temperature of 40° F., gives the theoretical velocity in ft. per min. for various heights between inlet and outlet and inside temperature.

TABLE LXXV

Outdoor Temp. 40° F.	Theoretical velocity in ft./min. for height between inlet and outlet.							
Indoor Temp.	5 Ft.	10 Ft.	15 Ft.	20 Ft.	30 Ft.	40 Ft.	50 Ft.	100 Ft.
°F.								
50	108	152	188	218	265	307	344	485
55	133	188	230	266	325	376	420	590
60	152	214	262	303	370	428	480	675
65	168	238	290	336	410	475	530	750
70	185	260	318	366	450	520	580	825
75	195	276	338	391	478	552	615	875
80	208	294	360	416	506	586	655	930
85	221	311	380	441	538	621	695	985
90	233	332	403	466	571	656	735	1050

The theoretical velocity is not obtained in practice, due to resistance to air flow. The practical flow is commonly taken at ½ to ⅔ of the theoretical. The effect of wind is to increase the velocity, but this depends on the type of roof outlet.

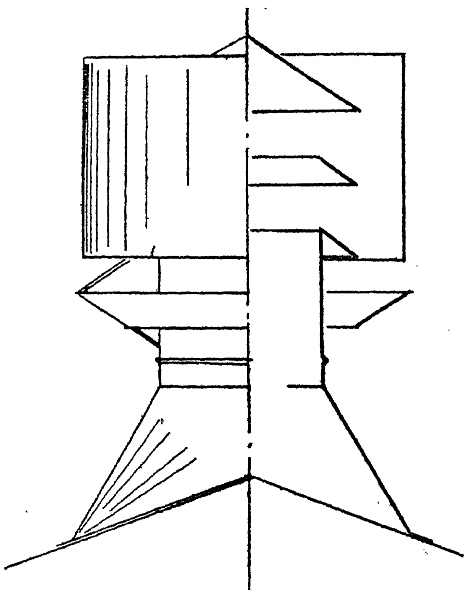


FIG. 224.—Robertson's Ventilator.

One well-known type of roof ventilator is the Robertson shown in Fig. 224. The exhaust capacities for these are given in the makers' catalogue, from which the following table is abstracted:

TABLE LXXVI

EXHAUST CAPACITIES OF ROBERTSON VENTILATORS, CU. FT. PER MINUTE PER VENTILATOR

Diam. of Ventilator	Temp. Diff.	Height of Ventilator above Intake. Ft.	Wind Velocity. Miles per Hour	
Ins.		—	4	10
12	10°	20	216	361
		40	267	411
	30°	20	304	449
		40	391	536
18	10°	20	489	816
		40	602	929
	30°	20	686	1,013
		40	882	1,209
24	10°	20	868	1,449
		40	1,069	1,650
	30°	20	1,219	1,800
		40	1,567	2,148
36	10°	20	1,952	3,259
		40	2,405	3,712
	30°	20	2,743	4,050
		40	3,521	4,828
48	10°	20	3,471	5,793
		40	4,276	6,598
	30°	20	4,878	7,200
		40	6,263	8,585
72	10°	20	7,809	13,029
		40	9,619	14,839
	30°	20	10,974	16,194
		40	13,986	19,306

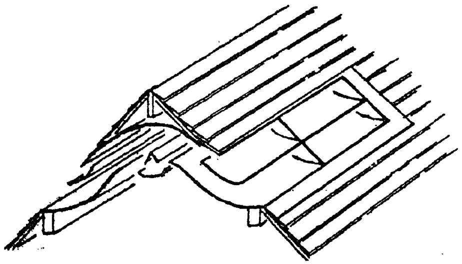
Intermediate sizes are 30″, 54″ and 60″ diam.

If N = Number of roof ventilators,
V = Volume of air per ventilator per minute from above table,
Ch = No. of airchanges per hour in room,
C = Cube of room,

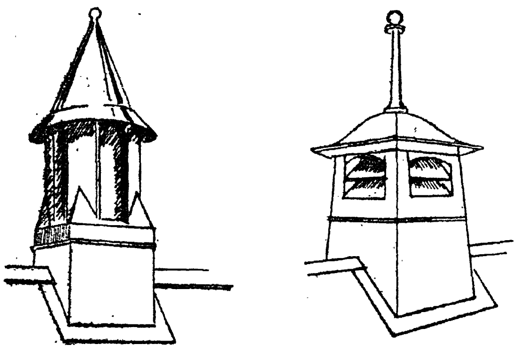
$$N = \frac{C \times Ch}{V \times 60}.$$

In order to prevent undue loss of heat through roof ventilators in cold weather, a damper may be provided in the base operated by cords or rods from floor level.

Some other conventional forms of roof extractors are shown in Fig. 225.



'Concealed' Type of Roof Extractor.



Typical Extractor Turrets.
FIG. 225.

For factories, furnace buildings, etc., in which much heat or steam arises from the process carried on, a 'Jack' roof type of construction gives a considerably greater extract area than a multitude of single ventilators, and at less cost. It does not function satisfactorily in wind, but can be made to do so by the provision of baffles as shown in Fig. 226. Correspondingly large inlet openings must be provided near the floor for the admission of the requisite volume of fresh air.

Air Inlets behind radiators have been referred to earlier (p. 180). The free area of inlet gratings should be the same as the extract, or based on a velocity of 300 to 500 ft. per minute, whichever is the greater. Convectors may similarly be fitted with fresh-air inlets. The 'flue effect' of a convector or indirect radiator at the base of a rising duct can on its own account induce a considerable air flow (Fig. 227). The higher the flue the greater the velocity, and the greater the emission from the heater (see reference to convectors on p. 179 and p. 326). The entering air deposits dirt and dust on the heaters, which are not visible and hence forgotten,

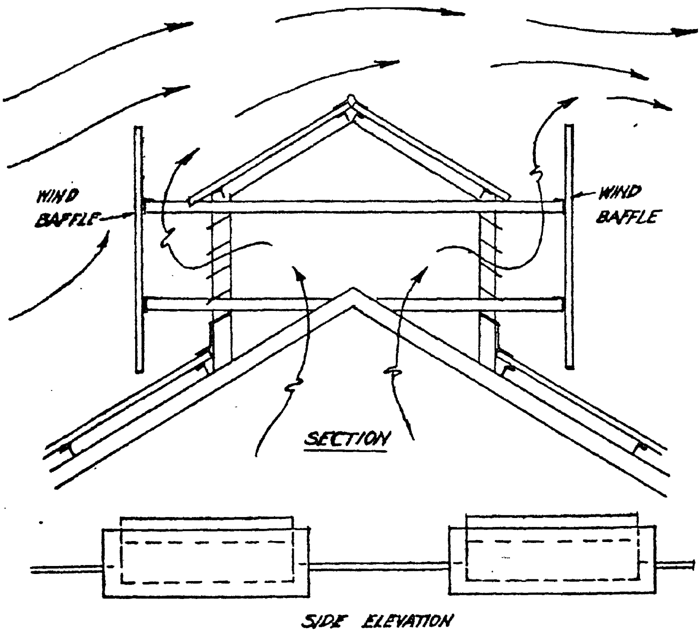


FIG. 226.—Extraction by Jack Roof Ventilation.

and for this reason this type of heating-cum-ventilation is not to be recommended.

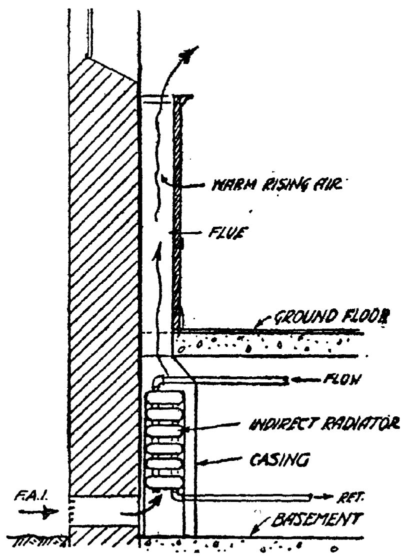


FIG. 227.—Ventilation induced by Indirect Radiator.

Air inlets by 'Tobin Tubes' (Fig. 228) are often seen in old buildings, but are generally to be found choked with dirt or blocked up on account of draughts. The latter may be said to be the objection to most wall gratings and other devices designed to admit fresh air without first warming it.

Tobin tubes were frequently combined with roof extractors in which a strong rising current was induced by gas jets at the base, as shown in the

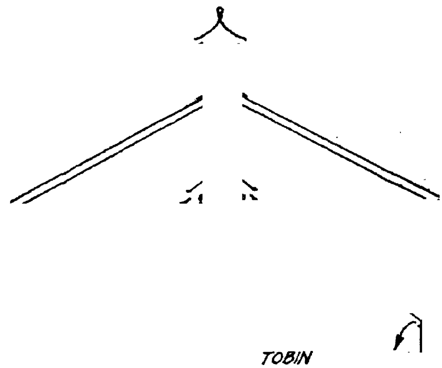


Fig. 228.—Old Method of Ventilation by Tobin Tubes and Gas Jet Ventilation.

figure. This was one of the few practicable ventilation systems available before the days of electrically-driven fans.

(2) **Natural Inlet and Mechanical Extract** (Fig. 229)—The system func-

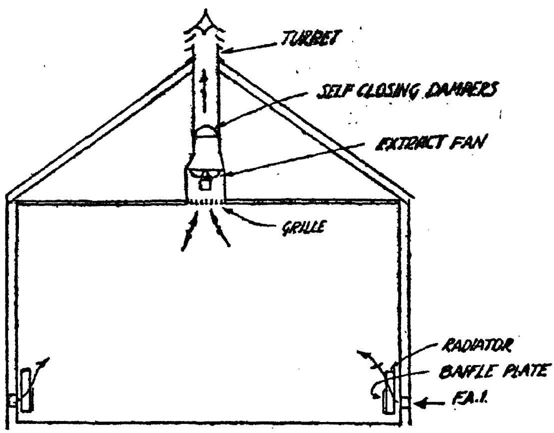


Fig. 229.—Ventilation by Natural Inlet and Mechanical Extract.

tions irrespective of wind and temperature differences, and is positive in action. It is often suitable for workshops, factories, assembly halls, etc. Owing to the negative pressure set up in the room there will be a tendency

to inward leakage rather than outward, thus preventing escape of steam and smells, etc., to other parts of the building, and it is thus particularly suitable for laundries, kitchens, lavatories, laboratories, and rooms where fumes or noxious vapours are given off from process work.

The inlet may be by fresh-air inlets behind radiators or heating coils

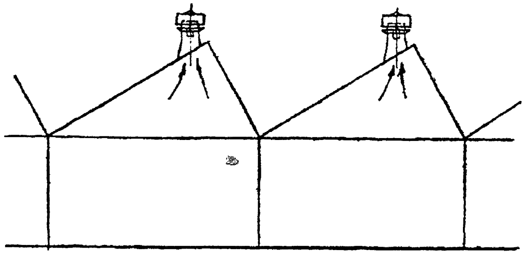
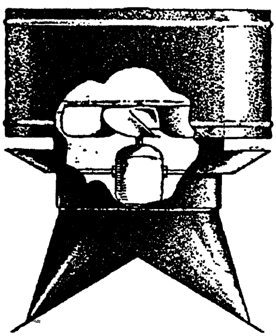


FIG. 230.—Showing Fig. 230 (*a*) as applied to North Light Factory.

as for (1) above. When doors and windows are shut the suction produced by the fan will cause a positive pull through the grating and over the radiator or coil.

The extraction may be by means of propellor fans. Where discharging directly through roof, as in a factory, a convenient type is shown in Fig. 230. For use in a vertical or horizontal duct the arrangement may be as in Fig. 231.



CAPACITY TABLE

	FAIRLY QUIET			QUIET			
NOMINAL SIZE	VOL C.F.M.	REVS.	WATTS ABSORBED 3 PHASE	VOL C.F.M.	REVS	WATTS ABSORBED 3 PHASE	NETT WEIGHT IN LBS.
12	1150	1400	85	750	900	55	50
15	2200	1400	145	1400	900	63	75
18	2450	900	98	1900	700	76	112
24	4300	700	155	3300	550	140	185
30	6500	550	290	5400	460	180	320
36	11000	550	570	8000	400	320	490

FIG. 230 (*a*).—Combined Roof Ventilator and Extract Fan.
(Positive Flow Ventilators, Ltd.)

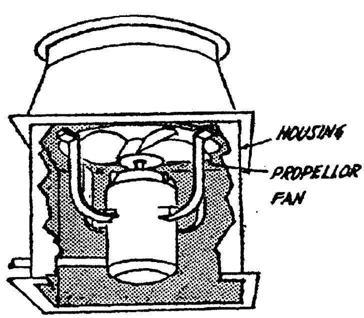


FIG. 231.—Extract Fan in Duct Housing (Wood's).

Where exhausting through a wall, the fan would be fixed as in Fig. 232, and should preferably be on the side of the room remote from the inlets.

The effect of wind blowing in opposition to a fan discharging through a wall opening is to hold up the delivery of air, or to make it hunt, produc-

ing a surging gusty action, often with some noise, and with a considerable reduction in volume exhausted. If it is possible to turn the discharge vertically upwards this is preferable, as the wind then aids the extraction. If this

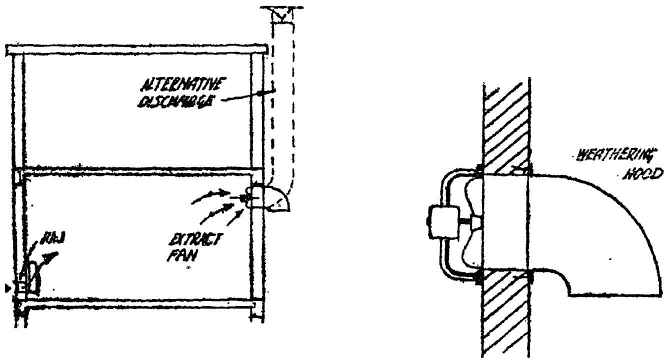


FIG. 232.—Wall Extract Fan.

is not possible a baffle placed in front of the fan discharge will improve the operation. This is shown in Fig. 235.

To prevent the escape of warm air when the fan is shut off, self-closing dampers may be fitted (Fig. 233).

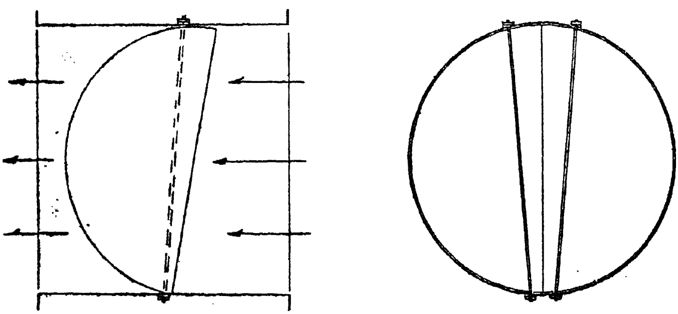


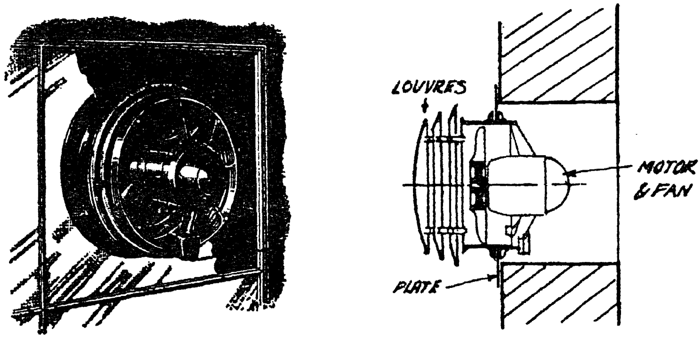
FIG. 233.—Butterfly Self-closing Dampers.

For small rooms one of the moulded plastic type window or wall fans shown in Fig. 234 will frequently give a useful solution of a ventilating problem.

In the case of a larger hall, or where a number of rooms has to be dealt with, an extract duct system becomes necessary (Fig. 235). The fan must then be capable of overcoming the resistance of the ducts, and the ordinary propellor fan then tends to become noisy. Use may then be made of a cased fan (p. 506) or of an 'Axial Flow Fan' (p. 509).

Sizes of fans can only be selected from makers' data. It is not possible to give a summary here as the range for each size and type is so wide.

The arrangement of ducting may be as simple or as extensive as is



Mounted in Window. Mounted in Wall.
Fig. 234.—Small Extract Fans (Vent Axia).

necessary to meet the case. A typical arrangement for a kitchen is shown in Fig. 236. Here it will be noted that hoods are provided over the main fume-producing items of equipment. For a group of lavatories it would be arranged somewhat as in Fig. 237.

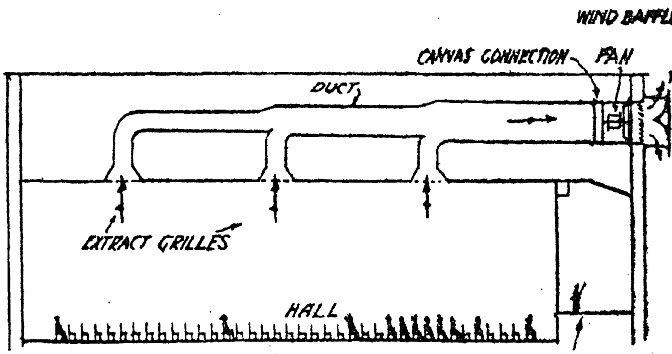


Fig. 235.—Ducted Extract System to Hall.

Internal Lavatories—A special case occurs in connection with the method of planning lavatories entirely within the building with no opening direct to outside.

Public Health Authorities, where such an arrangement is permitted, lay down a standard. The London County Council, for instance, calls for 750 cu. ft. per hour per soil pan or basin; this is, however, a very low figure. In practice more is required. Six changes per hour is generally accepted as a minimum.

The regulations further state that exhaust fans must be in duplicate. This involves an arrangement such as shown in Fig. 238.

Provision must be made for inlet air by suitably placed fresh-air inlets. In a block of flats, for instance, where lavatories are vertically above one

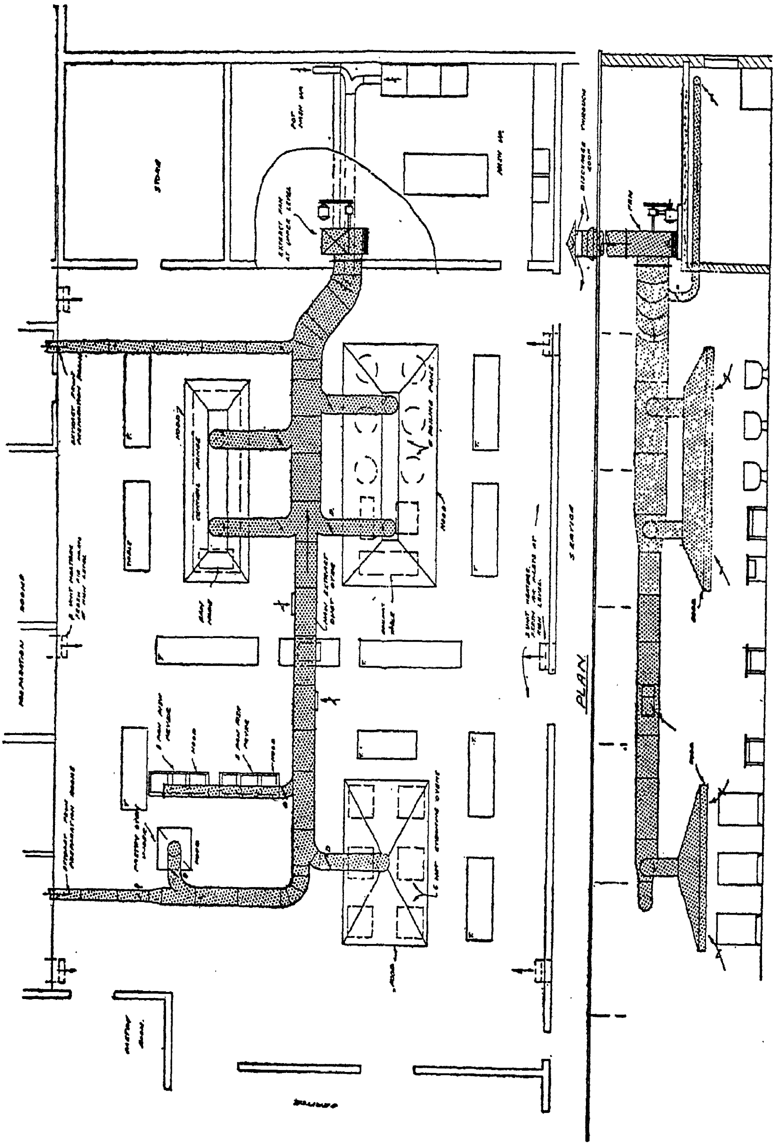
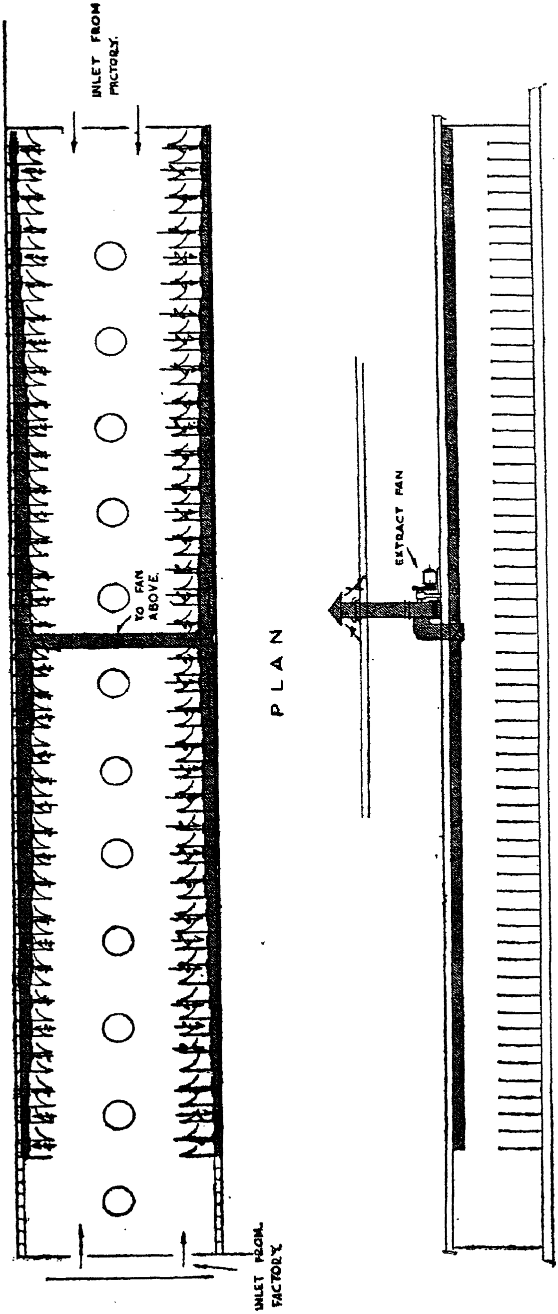


FIG. 236.—Plan and Section of a Typical Kitchen Extract Ventilation System.



Lavatory Block.

another, these inlets may connect to a rising shaft communicating with outside air at the bottom, where a heater may be placed. The shaft may form the duct for the various pipes serving the lavatory fittings.

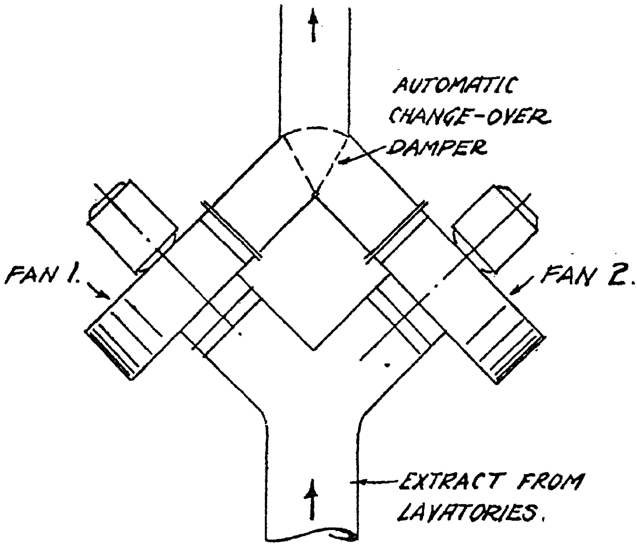


FIG. 238.—Duplicate Fans for Internal Lavatories.

(3) **Mechanical Inlet and Natural Extract**—This method is suitable for certain types of factory, offices, boilerhouses, etc.

In the case of a factory the inlet may take the form of a series of fresh air unit heaters (p. 323) with extract by natural roof ventilators.

In the case of offices and similar rooms a ducted system for the inlet

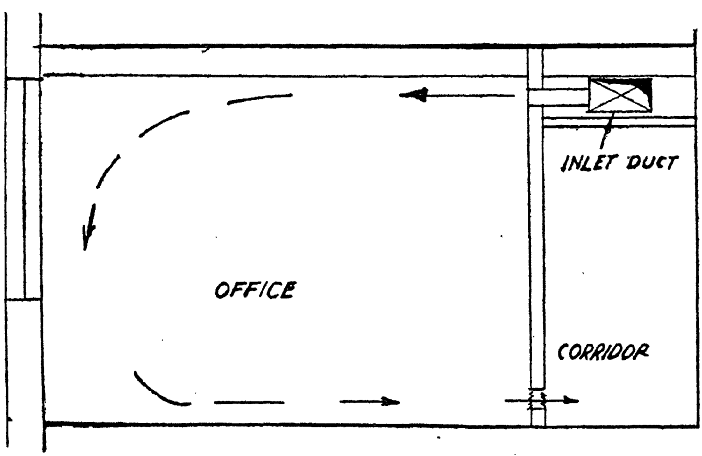


FIG. 239.—Ventilation by Mechanical Inlet and Natural Extract.

may be used, delivering the air into the room at high level, and with louvred openings at low level, allowing the extract air to pass out into the corridor (Fig. 239). Provision must be made for the corridor to connect to a space of free escape, such as a staircase connecting to the street, or to a natural vent shaft open at the top.

Small single offices may be served by a small cabinet type ventilating unit, one type of which is shown in Fig. 240. This has provision for filter-

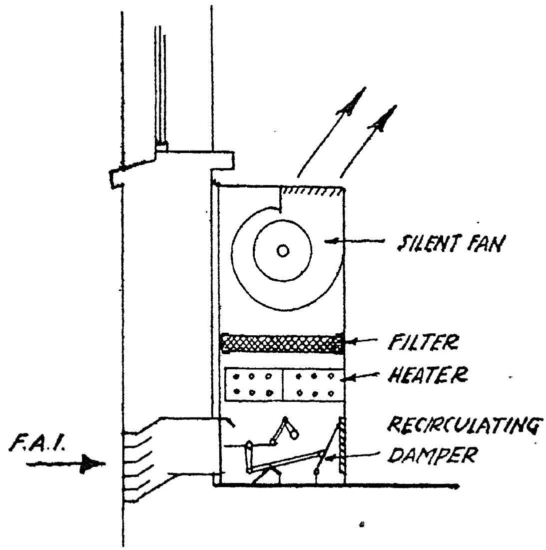


FIG. 240.—Cabinet Type Unit Ventilator.

ing, heating and humidifying, and may be set to introduce fresh air or recirculate. If provided with cooling coils it becomes a Unit Air Conditioner, referred to on p. 446.

Ventilation of Boilerhouses—Boilerhouses are a special case owing to the air consumed in combustion. In any boilerhouse provision must be made for this air to enter. The volume may be estimated in the manner given under 'Combustion'. Where a boilerhouse is above ground this air often provides adequate ventilation without augmentation. In the case of a confined basement boilerhouse, additional air supply may be necessary to keep the temperature down to a reasonable limit. The amount of heat liberated and volume of air necessary can easily be calculated.

It is then preferable to arrange for this air to be provided by mechanical inlet rather than by mechanical extract, since the latter may tend to create a negative pressure counteracting the effect of the flue. With mechanical inlet, provision must be made for the surplus extract air to escape by natural means.

(4) **Mechanical Inlet and Extract**—This method of ventilation is capable

of the widest application, as distribution and pressure are both under control, as is also temperature and humidity within the limits already referred to. Air filters may be included for the cleaning of the air. It is particularly applicable to theatres, cinemas, restaurants, dance halls, banqueting halls, smoking rooms, libraries, hospital operating and treatment rooms, offices, canteens, basement kitchens, etc. Fig. 241 shows a diagrammatic arrangement.

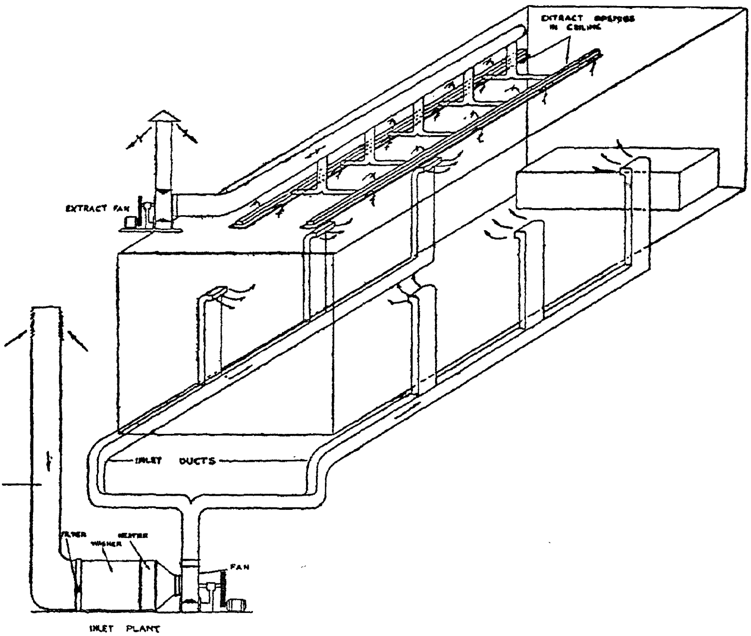


FIG. 241.—Mechanical Inlet and Extract System.

In all these cases both inlet and extract would most probably require to be served from fans at a distance by means of ducts.

In normal living rooms, offices, etc., where no fumes or smells are generated, the extract should be less than the inlet, so that draughts are outward not inward; it is then usually - -	75 to 90 per cent. of inlet.
Where fumes arise, and a negative pressure in the room is necessary, as in hospital treatment rooms, kitchens, shops, etc., the extract should exceed the inlet, usually taken at about - -	110 to 120 per cent. of inlet.

There are a great number of combinations of inlets and extracts with different kinds of fans, with or without ducts, etc., to suit various purposes,

but enough has no doubt been said to indicate the general principles and possibilities.

DESIGN OF INLET SYSTEM

The design of the inlet ventilation system with ducts is dealt with in Chapter XIX. Fans, heaters, filters, air-washers, controls, ducts, may all be similar to those described therein.

The method of distributing the air will follow one of the systems mentioned in Chapter XIX. As the volumes with straight ventilation are generally less than is the case with air-conditioning, the upward system is more general than the downward.

Heat for Air—When inlet air is introduced at about room temperature, the heating system is relieved of the duty of warming infiltration air because leakage tends to be outward. Radiators, heating panels, etc., should therefore be sized on the fabric losses only. This leads to obvious economy and avoids over-heating, which otherwise occurs.

Where air flow is induced naturally over radiators or coils, the heat to raise its temperature to that of the room (e.g. from 30° to 60°) must be supplied by this apparatus, and the design must allow for it accordingly. The B.T.U.'s per hour required from the heating system will equal weight of air passing per hour × temperature rise × specific heat (·24).

The weight is arrived at by dividing the volume per hour by the volume in cubic feet per pound from Table LXXXI (e.g. at 60° = 13·29).

The increased emission from radiators due to ventilating effect is referred to on p. 180.

Where the inlet is mechanical the warming of the air will be by means of a heater battery, the heat supply to which will be determined in a similar manner to the above. Some allowance should be made for drop in temperature in ducts if these traverse unheated spaces.

If an air washer or humidifier is provided, the heat for evaporation of the moisture gain must be provided in addition, either by a spray water heater, or by an air pre-heater. The method of calculation of the heat necessary is given on p. 465.

PLENUM SYSTEM

The name 'Plenum System' is generally understood to mean a system of ventilation in which sufficient heat is put into the inlet air, not only to raise it to the desired room temperature, but to a higher temperature, so that the warmed air will balance the radiation and other heat losses of the building.

This generally involves bringing in the air at a temperature of 80° to 120° (depending on airchanges, exposure and proportion of windows, etc.), and it tends to produce a feeling of stuffiness.

Its only advantage is that it saves the expense of a separate heating system, but unless recirculation of the air is arranged for it is extravagant in fuel, and even with recirculation, it saves no fuel and necessitates the running of fans continuously to maintain heat.

It has been used in banks and office buildings, but is chiefly used in factories, hangars, sheds, etc. Its use for heating a large building with many small rooms involves many difficulties.

For large open spaces, such as large factories, it is an alternative to the unit heater system, in which the air is heated at a number of points and is delivered direct, not through ducts. It is debatable which of the two systems is to be preferred for this case. The Plenum system is simple, it avoids a multitude of small units requiring maintenance. The Unit Heater System avoids ducts, gives greater ease of temperature control and better distribution of heat. Each system has its application, but Unit Heaters are, on the whole, tending to displace the Plenum System. Unit Heaters are only suitable where some noise does not matter, while a Plenum System can be almost silent.

Design—The following are a few notes on design. They may be read in conjunction with Fig. 242.

(1) The fresh-air inlet should be from a point not contaminated by fumes.

(2) The inlet air should be filtered, or heaters and ducts will become

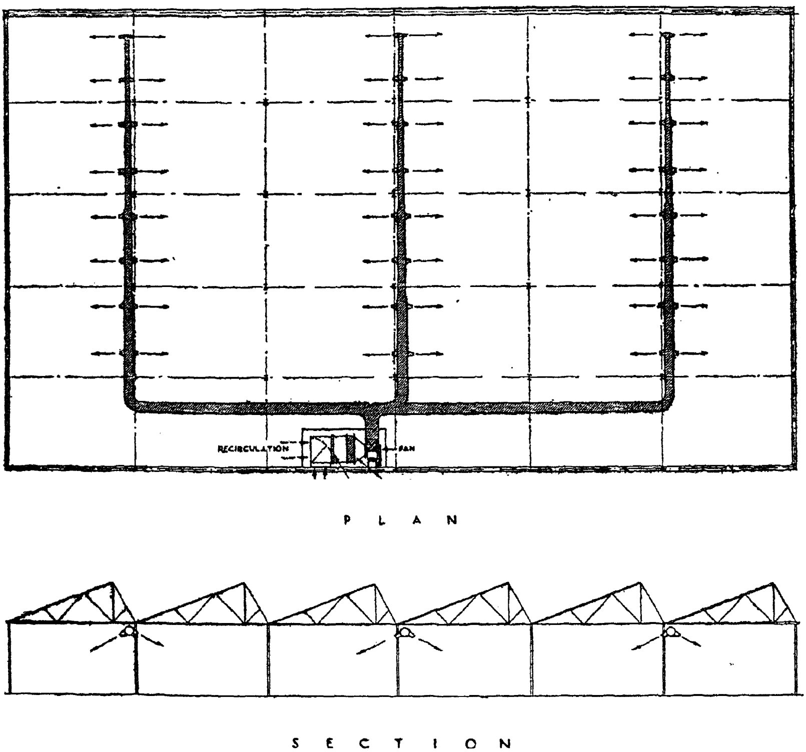


FIG. 242.—Typical Lay-out of Plenum System.

quickly coated with dirt. The walls and ceilings of the building to be heated will also get very dirty. An air-washer may be used if desired.

(3) The heater battery may be heated by hot water, steam, gas or electricity.

(4) Thermostatic control is desirable to prevent over-heating. It may take the form of a room-stat and duct-stat, controlling the admission of heat to the heater battery. The duct-stat acts as a low limit to prevent admission of air below a pre-determined temperature. It may, alternatively, with hot water or steam, be simply a direct acting thermostat giving a constant air outlet temperature.

(5) The fan will be of the cased type to give the necessary pressure to overcome the resistance of the ducts, heater, etc., generally between $1\frac{1}{2}$ and $2\frac{1}{2}$ in. w.g. It may be of constant speed type, unless an increased volume is required in summer, when it may be of 2-speed or variable speed type.

(6) Means for recirculation of the air should be provided for quick heating up, and to economize fuel in winter.

(7) The final air temperature should not exceed 110° to 120°. Temperatures above this cause difficulties of distribution as in the case of Unit Heaters.

(8) The rate of air delivery will be determined from the heat losses of the building. Thus:

$$\text{lbs. air to be delivered per hour} = \frac{\text{Heat loss in B.T.U./hr.}}{\cdot 24 \times (\text{Final temp.} - \text{room temp.})}.$$

If the air volume delivered is equal to about 3 to 4 changes per hour, there should be no difficulty in giving good distribution.

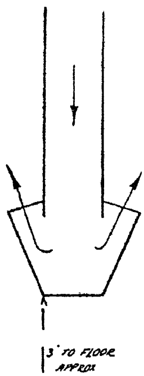
(9) If 100 per cent. recirculation is always used, the heat input to the air heater will equal the heat losses. If, however, the system can admit 100 per cent. fresh air, even in 30° weather, the heat input from 30° to room temperature will be additional. This makes the system very wasteful as, if 60° is the room temperature and 120° the air temperature as delivered to the room,

heat input is proportional to $120 - 30 = 90°$,
heat wasted ,, ,, ,, $60 - 30 = 30°$.

In other words, one-third of the heat input is lost, but this may be unavoidable if the ventilation so afforded is a necessity.

(10) Ducts are usually of circular metal slung overhead, with plain outlets or nozzles delivering at an angle to the floor. Some designers favour bringing the air down near floor level. With this arrangement it is difficult to avoid draughts, and the form of outlet shown in Fig. 243 has been developed to overcome this. Large main ducts should be insulated. Alternatively, ducts may be placed in the floor, in which case they should all be insulated. Branch ducts will then rise up from the floor terminating

with a suitable outlet. Duct velocities should be high (say, 2,000 to 2,500 ft. per min.), to reduce heat loss from duct surfaces.



(11) A margin of, say, 25 per cent. in heater battery power is desirable to reduce warming-up time and to allow for losses in ducts.

(12) Caution should be exercised in serving several rooms or workshops from one central plant, if they are of widely differing heat loss and occupancy.

A separate heater battery may be furnished for each section with separate room thermostats, all being served from the same fan.

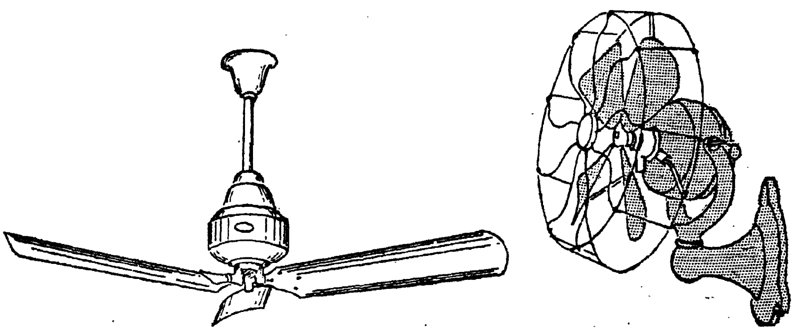
(13) It is not usual to provide extract fans where a Plenum System is installed. Leakage should be outward not inward, and the normal leakage from the building is usually sufficient as a means of egress for a sufficient airchange.

(14) The effect of strong wind on a large space heated by Plenum System is sometimes to cause the warm air to 'pile up' on the leeward side. There is no cure for this, except to stop air leakage, which may be impossible.

(15) A Plenum System may be run in summer without heat, to give ventilation.

PUNKAH OR DESK TYPE FANS (FIG. 244)

Fans of this type merely stir round the air of the room, and in so doing cause a rapid movement over the skin with resultant evaporation and cooling, which accounts for their widespread use in the tropics.



Punkah Fan. Desk Fan mounted on Wall Bracket.

FIG. 244.

They are useful in giving relief in crowded ill-ventilated rooms, and in offices, shops and the like in hot weather. With an adequately designed ventilating system they should not be necessary.

CHAPTER XVIII

Air-Conditioning

The science of air-conditioning may be defined as that of supplying and maintaining a desirable internal atmospheric condition irrespective of external conditions. As a rule 'ventilation' involves the delivery of air which may be warmed, while 'air conditioning' involves delivery of air which can be warmed or cooled and have its humidity raised or lowered.

The application of the science is divided into two categories: industrial air-conditioning for processing of materials, and air-conditioning for human comfort and efficiency. It is to this latter application that the following chapters refer. Broadly speaking, there are four items in comfort air-conditioning which require analysis, namely, the heat content or temperature; the moisture content or humidity; the purity; the distribution.

The desired atmospheric condition usually involves a temperature of 60° to 70° F. in winter and 70° to 75° in summer; a relative humidity of about 50 per cent. to 60 per cent.; and a high degree of air purity. This requires different treatments according to climate, latitude, and season, but in temperate zones such as England it clearly involves:

In Winter—A supply of air which has been cleaned and warmed. As the warming lowers the relative humidity, some form of humidifying plant, such as a spray washer with preheater and main heater whereby the humidity is under control, is generally necessary.

In Summer—A supply of air which has been cleaned and cooled. As the cooling increases the relative humidity, some form of dehumidifying plant is essential. This dehumidifying is generally accomplished by exposing the air to cold surfaces or cold spray, whereby the excess moisture is condensed and the air is left saturated at lower temperatures. The temperature of the air has then to be increased, to give a more agreeable relative humidity. This can be done by warming or by mixing with air which has not been cooled.

Dehumidifying can also be brought about by passing the air over certain substances which absorb moisture. Thus, in laboratories, a vessel is kept dry by keeping a bowl of strong sulphuric acid in it or a dish of calcium chloride, both of which have a strong affinity for moisture. Silica-gel, a form of silica in a fine state of division exposing a great absorbing surface, is used also for drying air on this principle, but this process has its difficulties, and is not generally used in comfort air-conditioning applications in this country.

THE CASE FOR AIR-CONDITIONING

It is often seriously contended that there is no case for true air-conditioning, with means for cooling, in a climate like that in England.

It is not, of course, denied that there are many buildings where the necessity is not absolute, or that custom has not made us willing to accept, in hottest summer, conditions of extreme discomfort, while a similar divergence from comfort conditions in the opposite direction in winter would never be tolerated. This is partly because we are used to looking on an adequately warmed room as essential, and partly because the cost of warming is much less than the cost of cooling. The question of the necessity for air-conditioning thus becomes one of how great discomfort can be tolerated, and what can be afforded. The question of air-conditioning should certainly be considered very carefully, when planning any buildings whose prime function is to attract paying customers, e.g. restaurants, hotels, cinemas, department stores, office blocks, exhibition halls, etc.

One important factor is the degree of concentration of people which may occur in the building. The effect of this concentration is not always understood, but perhaps the following example may make it clearer.

The authors had occasion to air-condition a banqueting hall, 73 ft. by 50 ft. by 25 ft. high, containing 400 occupants, lighted (even in daytime) with electric lamps aggregating 7800 watts. The heat released in the room in B.T.U.'s per hour may be calculated as follows:

	B.T.U. per Hour
1. *By occupants 400 × 300 B.T.U. - - - -	120,000
2. †By electric lights 7·8 k.w. × 3415 - - - -	26,800
3. By heat transmission through walls and roof from outside and infiltration - - - - - -	31,000
Total - - -	177,800

The cubic content was 91,700 cu. ft. and the air necessary was estimated at 720,000 cu. ft. per hour (approximately 8 changes per hour) so that the heat released would raise the temperature by

$$\frac{177{,}800}{720{,}000 \times 0{\cdot}019} = 13^\circ \text{ F.}$$

This air delivery is at the rate of 1800 cu. ft. per hour per occupant, or nearly double that called for in the L.C.C. theatre regulations, which require 1000 cu. ft. per occupant. The introduction of such a degree of air-change into a room without causing draughts is one of the chief problems of air-conditioning. Yet if the volume is much less and the temperature rise much more than 13° F., difficulty is experienced in getting the incoming cool air to mix, and cold currents result.

* See p. 42. † See p. 43.

In England we occasionally have shade temperatures over 90° F. in summer and often over 80° F.; and in the twelve months ending November 1933 there were, in London,

192	days when the temperature was below	60° F.
89	,, ,, ,, ,, ,, ,,	60–70° F.
84	,, ,, ,, ,, ,, ,,	over 70° F.

If we add 13° F. to these temperatures for the rise in the hall, it follows that without refrigeration or cooling there would be:

192 days when conditions were comfortable (i.e. under 73° F.).
89 days when it would be uncomfortably hot (73° to 83°).
84 days when it would be unbearable (over 83° and rising to 93° when the outside temperature is 80° F.).

Had there been only 40 people in a hall of the same size instead of 400, the heat released would only be 69,800 B.T.U. per hour (i.e. one-third), so the temperature would be only about 5° (with the same air volume). But we should then be delivering 18,000 cu. ft. of air per hour per occupant and the result would be too fresh for comfort. Some form of control is therefore necessary to adjust for such variation of conditions, automatically reducing the air volumes or raising their temperature or both.

Had the same floor area been crowded, as in a cinema, the possible occupants would be 800, and the temperature rise would reach 22° F. approximately, making the internal temperature over 90° F. on the 84 days when the external temperature is over 70° F.

When these very high temperatures are reached the inward heat transmission through the building and the infiltration loss disappear, and the sensible heat released per occupant decreases until at 98·4° it ceases altogether. The difference is, of course, made up by the latent heat of the perspiration, or water vapour. Thus it is doubtful whether a temperature approaching 98·4° F. would ever be attained. But if anyone derives much comfort from this thought, they are surely entitled to it. To us it would be cold comfort—though perhaps 'cold' is not a happily chosen word.

Anything much above 70 to 75° F. is generally considered outside the range of comfort conditions in summer, particularly if it is accompanied by a high relative humidity (see Fig. 15 and the accompanying notes). Hence the case for refrigeration is proved wherever concentration of people occur.

Let us now consider the case of important city offices. The installation of air-conditioning enables the following advantages to be obtained:

(*a*) The use of basements as fully occupied offices with light and ventilation as perfect as above ground (and in summer often more so). This greatly increases the value of a building on a site of limited size.

(*b*) Windows can be kept closed, and can indeed be double, so excluding the noise of outside traffic which often makes concentration and telephoning impossible, and also the dirt invariably associated with city air is eliminated.

Until a few years ago, the owners of New York skyscrapers, which are usually let off as offices, found that the first to the fifth floors were practically unlettable, owing to street noises and dirt which entered when the windows were open. The installation of air-conditioning systems has increased the renting value of these floors far in excess of those on the higher level.

It is not without interest that the new Bank of England is air-conditioned throughout so as to obtain these advantages. Another example which has been similarly treated is Princes House, also in the City, this being particularly interesting as it represents the conversion of an old building to modern standards, it being found that conditions with open windows were rapidly becoming intolerable.

Complete air-conditioning of buildings will have an increasing field of application as its merits become better appreciated. At present it is in its infancy in this country.

In the United States it has been found that air-conditioning has a money value, particularly in theatres, shops and restaurants where, without it, customers simply do not come in hot weather. Similarly in office buildings the output of work has been found to fall off so much in summer that the installation soon pays for itself. Flats, houses, trains and steamships are being fitted with plants with beneficial results.

Admittedly in this country our conditions are less severe, but where plants have been installed they have quickly proved a necessity rather than a luxury. Theatre managers bemoan the fact that they cannot fill their theatres in summer, yet how many have tried the obvious remedy? It is not to be expected, in these days, that people will sit for three hours or so in the appalling conditions which still obtain in most of our places of entertainment. In the personal experience of one of the authors, 92° was reached in one old London theatre.

In the restaurants, the same applies. Most people lose their appetite in a hot stuffy atmosphere and the result is obvious. A number of proprietors have put in air-conditioning and cooling plants, and always find an increase in their takings.

As air-conditioning becomes more general, those who are without it will discover they are out of date and that they must either follow suit, or go out of business. So in the design of new buildings, particularly theatres, cinemas, hotels, restaurants and office blocks, the question of air-conditioning should be carefully weighed at the start, otherwise the enterprise may be behind the times before it is a few years old.

VARIOUS SYSTEMS DESCRIBED

Figs. 245 and 246 show two types of air-conditioning plant. The first is complete and is suitable for all types and size of building. The second is limited in capacity and is thus applicable only to comparatively small jobs (say up to 20 tons refrigeration capacity), such as single rooms, small restaurants, etc. It does not give such complete control over conditions as the previous plant, but the results are generally sufficiently near the ideal to be satisfactory. It is obviously cheaper in first cost.

A third system, shown in part in Fig. 247, is a combination of the two above, having direct expansion coils inside the air washer so that the water is sprayed over them, thus eliminating the separate evaporator coils. It is applicable to medium-sized installations, but a separate compressor is really necessary for each unit.

Complete Air-Conditioning—Dealing with Fig. 245, it will be seen that the fresh air represents only a portion of that supplied, the remainder being made up by recirculated air from the room. This reduces the amount of cooling or warming necessary since the return air has already been conditioned, at the same time allowing a low temperature-increment in the room. The amount of fresh air is controlled by that called for by regulations where these exist (such as the L.C.C. 1000 cu. ft. per hour per occupant) or otherwise by what is considered the safe minimum, generally about 600 or 750 cu. ft. per hour per occupant. This may only be a third or less of the total air supply.

The mixed intake is filtered through a 'dry' air filter, which may be of oil-coated or other type, as discussed later. The purpose of this is to remove the major portion of the dirt in the atmosphere.

Following this it passes through the air washer and dehumidifier. This is an enclosure containing one or more 'banks' of water sprays forming a fine mist; followed by a series of zig-zag plates known as the scrubber plates, over which a stream of water is maintained; followed again by the eliminator plates. These are intended to catch the free moisture entrained in the air stream, and allow it to return to the tank at the base of the washer for re-use.

The spray or mist regulates the amount of moisture in the air supply. In winter when the entering air is cold and dry, moisture needs to be added, and the spray is accordingly warmed in the heat exchanger on the pump delivery. The circulation through the cooler is then cut off and the water is simply drawn from the water tank by the pump and delivered back to the sprays via the heat exchanger. Alternatively a preheater is sometimes used for warming the air before entering the washer. The number of spray nozzles in use in winter is generally limited to one bank, and often a small separate pump is provided, with a larger one for summer use.

Two banks of spray nozzles, however, are more satisfactory, as this gives a greater volume of mist and assists towards better humidification of

the air. Apart from this, one bank would be inadequate when operating on summer dehumidification as the quantity of water handled would not be sufficient to limit the temperature rise to a reasonable degree.

As the water is evaporated into the air stream it is made up through a ball valve in the washer tank fed from the cold main.

The washer in winter thus acts as a humidifier.

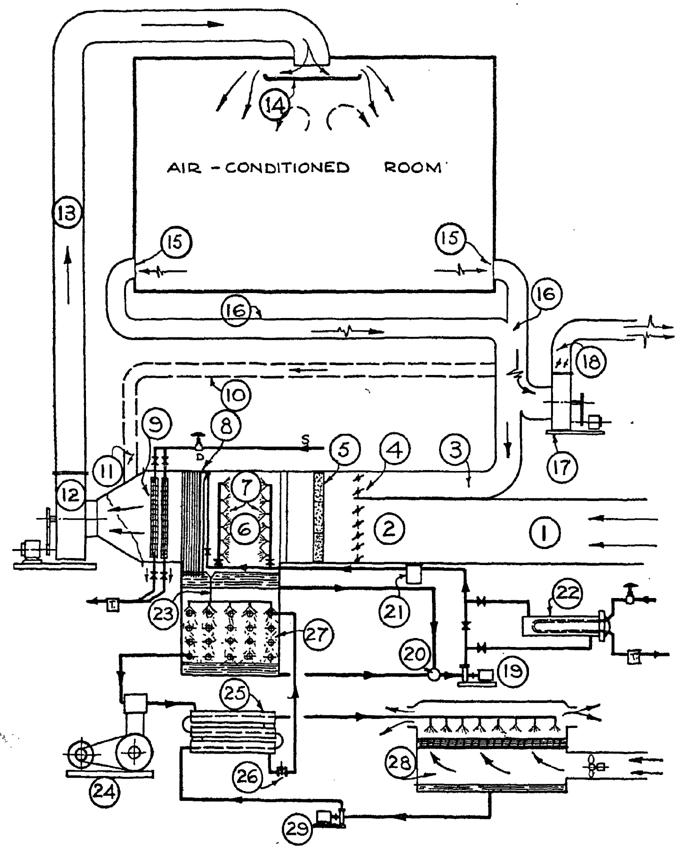


FIG. 245.—Diagram of a Complete Air-Conditioning System employing Air-Washer and Dehumidifier.

1, Fresh air intake. 2, Fresh air louvres. 3, Recirculated air intake. 4, Recirculated air louvres. 5, Air Filter. 6, Air washer and dehumidifier. 7, Sprays. 8, Scrubber and eliminator plates. 9, Heater batteries. 10, Recirculated air by-pass. 11, By-pass dampers. 12, Inlet fan. 13, Inlet duct. 14, Inlet diffuser in ceiling. 15, Extract gratings near floor. 16, Extract duct. 17, Extract fan. 18, Extract louvres and discharge. 19, Washer pump. 20, Automatic mixing valve. 21, Filter. 22, Spray-water heater. 23, Water outlet to cooling coils. 24, Refrigeration compressor. 25, Condenser. 26, Expansion valve. 27, Evaporator or cooling coils. 28, Atmospheric water cooler. 29, Cooling-water pump. '*S*' denotes steam supply. '*D*' denotes diaphragm valve. '*T*' denotes condensate trap.

In summer, when cooling is required, the sprays are supplied by the pump as before, but the suction is drawn from the cooled water tank as well as from the washer tank. The mixture is regulated by the mixing valve shown, so that the spray water arrives at the temperature necessary to maintain the desired conditions. In cooling the air below the dew-point it will have both its temperature and moisture content reduced, and the washer is then acting as a dehumidifier.

It should be understood that the spray saturates the air at nearly the same temperature as the water. Thus the air leaving the washer is always at the temperature of the dew-point (or nearly so) and the automatic

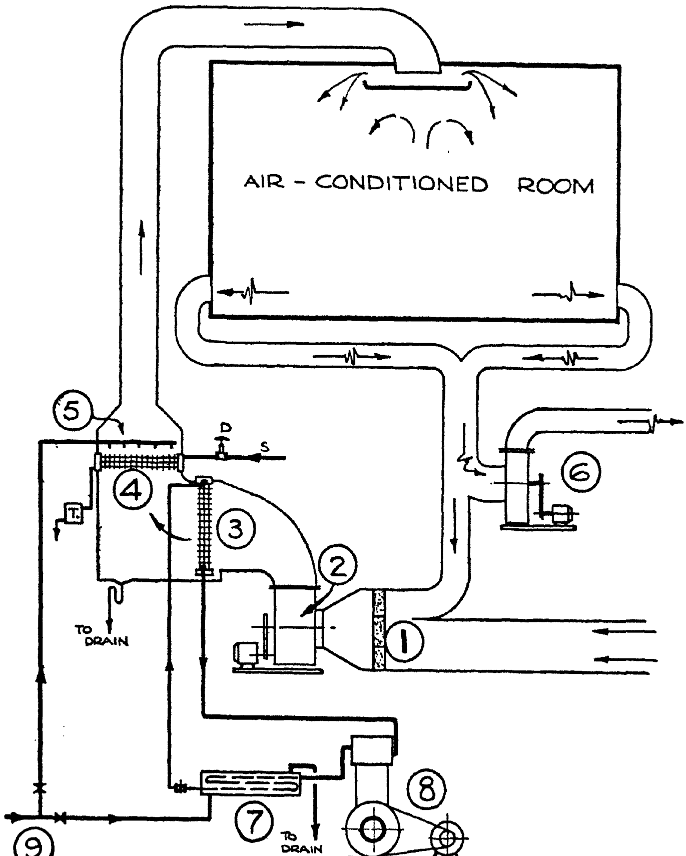


FIG. 246.—Diagram of an Air-Conditioning System employing Direct Expansion.

1, Air filter. 2, Inlet fan. 3, Direct expansion coils containing refrigerant gas. 4, Heating coil. 5, Humidifying sprays for winter use. 6, Extract fan. 7, Condenser cooled by main supply. 8, Refrigeration compressor. 9, Cold supply. Other parts not described are as Fig. 245.

control of the spray water temperature is arranged to keep this constant according to the season of the year. This is known as 'fixed dew-point' control. In winter the control is effected on the steam supply to the water heater and in summer on the cooled water inlet to the mixing valve.

Having established a fixed dew-point, the moisture content of the air is also fixed.

In addition to its function as a humidifier and dehumidifier the washer also assists in the cleaning of the air. The amount of dirt it collects depends largely on the kind of dirt encountered. Thus soot, which will not mix with water, is not brought down in a washer to any great extent, but dust and grit which have passed the dry screen will be caught in the water spray and carried on to the scrubber plates. These are constantly sprayed with water at the top so that any dirt caught on them is carried to the tank at the bottom, from which it can be removed by sludging out once or twice a week.

As a cleaning device the washer is not a great success, but that it does remove something from the air is apparent from the dirty state of the water after a few days' running. Some makers omit the scrubber plates as they have ceased to regard the washer as an air cleaner, but opinion on this point varies.

After leaving the washer the air is warmed in winter by the heater to 55° or 65°, i.e. to the temperature at which it will be delivered to the room. This reduces the relative humidity to a figure depending on the dew-point setting. In summer the cooled and saturated air also requires to be warmed to reduce its relative humidity, and this is either done by using the heater, as in winter, or by mixing a proportion of return air, using the duct shown dotted on the diagram.

The air is delivered to the room in one of many ways referred to later, and, in passing through, is raised in temperature on account of the heat picked up from electric lights, occupants, and in summer from the walls, roof, glass, etc., which will be at a higher temperature outside than inside. The air will also be raised in moisture content owing to the moisture given off from the occupants, in breathing and perspiration.

The air extracted is generally limited to about three-quarters of the inlet so as to cause any draughts from doors, etc., to be outward rather than inward. If this proportion is the same as the recirculated volume, no extract fan is necessary, but where this is not so the extract fan as shown is required to remove the surplus and discharge it to atmosphere.

The refrigeration portion of the plant, it will be seen, is independent of the air supply, and only comes in contact with the water from the washer tank. This equipment may be one of several types using various refrigerants. The heat removed from the water together with the power input of the compressor is dissipated to the cooling water of the condenser. In order to save the cost of wasting this water to drain in large plants it is generally cooled in an evaporative cooler on the roof as shown in the figure. Only the make-up water due to evaporation then has to be supplied.

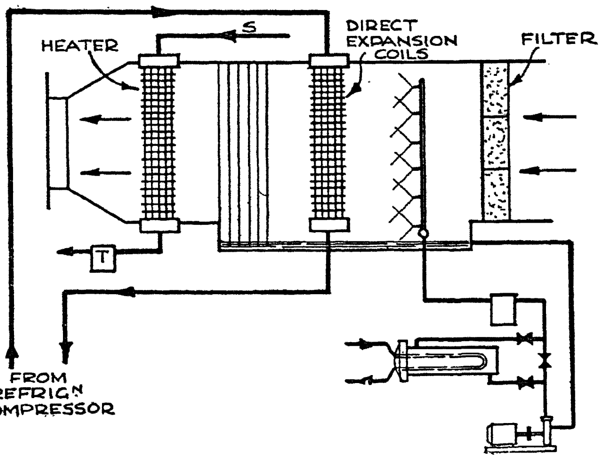


FIG. 247.—Direct Expansion Air-Conditioner Combining Air-Washer (for general description, see Fig. 245).

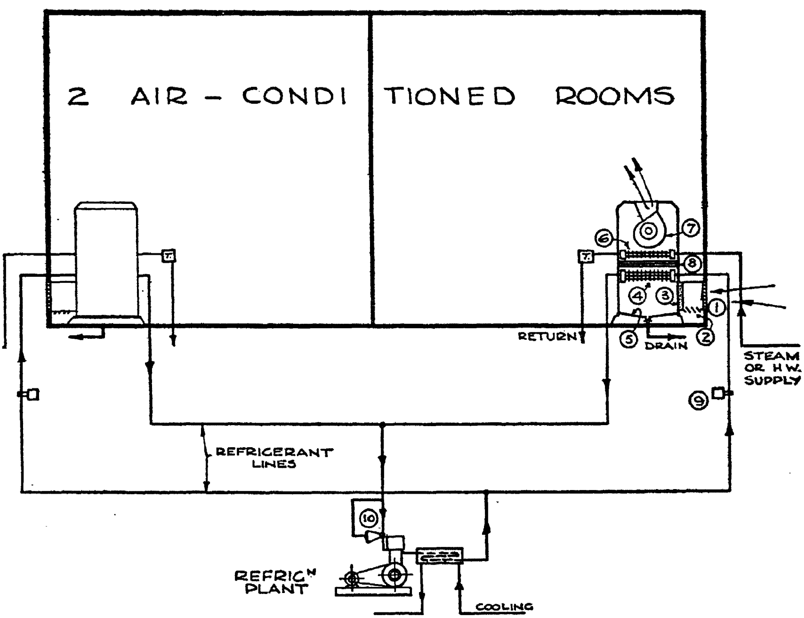


FIG. 248.—Unit Air-Conditioner.

1, Fresh air intake. 2, Recirculated air intake. 3, Filter. 4, Cooling and dehumidifying coils. 5, Drip tray and drain. 6, Heating coils. 7, Fan. 8, Eliminator plates. 9, Magnetic Valve, controlled from room thermostat. 10, Evaporator pressure control valve.

Air-Conditioning with Direct Expansion—Fig. 246 (p. 443) differs from the full system in that the refrigerant is passed through coils of gilled piping placed direct in the air stream, which become wet when dehumidification is taking place. With this method the gas used must be innocuous, as any leakage would quickly be discharged into the room. Control of temperature is effected by starting and stopping the compressor.

In winter the heating battery is brought into use, being supplied with steam or hot water, and, in order to raise the humidity, water from the main supply is dripped over the warm coils. The amount of humidification by this method can be under control.

Where more than one conditioning plant is provided, a separate refrigerator must be supplied for each, whereas with the previous system several units may be served from one machine. This, however, is not necessarily a disadvantage as the number of refrigerators in use at one time is the same as the number of plants, and if one or more rooms are not in use, then its plant is shut down, the others still working at full efficiency.

Direct Expansion System with Air-Washer—The type of system shown in Fig. 247 uses direct expansion coils containing the refrigerant, placed in the air stream inside the washer casing, and so arranged that the water is sprayed over them. This gives an economical system since the transmission rate of the wetted surface is very high, so reducing evaporator area, and in addition the water to be pumped is somewhat reduced.

Again it can only be used with refrigerants which are not objectionable when breathed, and there is difficulty in applying it to more than one plant with a single compressor.

Unit Air-Conditioners—A further type of air-conditioning system, which has been developed, is shown in Fig. 248 (p. 445). This system cuts out the necessity for ducting, there being a separate air-conditioning unit of small size in each room. The number of rooms which can be treated by this method can be anything up to ten or twelve. The individual conditioning units are all operated from a common compressor in just the same way as radiators are operated from a common boiler. The figure shows that the unit comprises a filter, cooling and dehumidifying coils, moisture eliminators, heating coils and silent circulating fan. The heat extracted from the room is led away in the refrigerant suction piping, which is usually in copper. Sometimes the same coils are used for cooling and heating.

Alternatively each unit may have its own compressor in the base, forming one combined unit.

The advantages of this system are that the space requirements for the whole plant are greatly reduced, although the control of temperature and humidity are not as perfect as with other systems. For office blocks, however, it has much to recommend it.

HEAT GAINS

The volume of air to be supplied for cooling purposes depends directly on the heat gain in the room and on the permissible temperature rise. The latter is limited by the difficulty of introducing air at very much lower temperature into a warmer atmosphere without draughts, and 10° to 13° difference is about the maximum.

The heat gains are made up of:

(*a*) **Conduction** through walls, roof, partitions, windows and doors. This may be estimated just as for ordinary heat losses for a heating system, taking 85° outside, 75° inside, i.e. 10° difference (or some other reasonable basis).

Theoretically the overall coefficients of heat transmission inwards in summer are not identical with the winter coefficients given in Chap. II, due to the different surface conditions. During the weather when the heat gain is likely to be greatest the air-movement outdoors will be very slight, and it can be shown that transmission coefficients will be reduced by the following amounts:

Coefficient '*U*'	Reduction
0·2	7%
0·3	10%
0·4	12%
0·6	17%
0·8	21%
1·0	25%

(*b*) **Sun Effect**—The solar radiation falling upon a building will heat up the inside in two different ways. Through glass it will pass directly as high temperature radiation, warming any objects upon which it falls, and these in turn will warm the air.

Radiation falling on opaque materials such as the walls and roof, will warm up the outer skin and heat will be transmitted through the wall by conduction.

Thus, in the case of glass the sun effect is superimposed on the normal transmission of heat by conduction due to difference of air temperature on the two sides. In the case of opaque materials, however, the conduction due to temperature difference is merged with the sun effect, the only result of the radiation being to augment the temperature difference between the two surfaces.

The amount of solar radiation falling on the surface depends on the

Latitude,
Orientation,
Exposure,
Time of day,
Season of year,
Atmospheric condition (haze, etc.).

Latitude determines the angle of elevation of the sun and hence its strength.

Orientation of a building determines the obliquity of the surfaces of walls, roof, etc., to the sun's rays.

Exposure is controlled by the height and presence of other buildings.

Time of day—As the sun moves round, different surfaces are exposed to it.

Season of year—The highest intensity of sun heat on a flat roof occurs in June, but on wall surfaces facing east or west the lower angle of elevation of the sun in spring and autumn may cause the maximum intensity to occur in these periods instead of being coincident with the maximum of the roof.

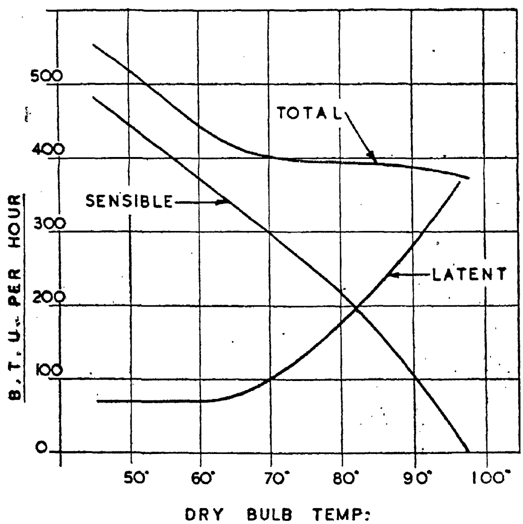
Atmospheric condition—Haze and smoke reduce solar intensity.

The method of computing the heat gain due to solar radiation is given in detail later (see pp. 458–461).

(*c*) **Infiltration** of hot outside air. This takes place even though a supposed pressure inside the building is created. Generally one-third or one-half air-change per hour is allowed in the calculations. Much can be done with double doors, etc., to check this loss. Windows must, of course, be kept shut for air-conditioning to be possible. It should also be remembered that the infiltration air brings in moisture which adds to the cooling load.

(*d*) **Heat from Occupants**—The heat emitted from the human body depends on activity (see p. 42), and on temperature, relative humidity and air-movement.

Fig. 249 shows for a man seated at rest, how the sensible, latent and



.249.— sensible, and total heat per hour for a man seated at rest. (From *A.S.H.V.E. Guide.*)

total heat varies with the dry bulb temperature for average humidities in still air. The sensible heat increases by about 20 per cent. with air-movements up to 100 ft. per min. The latent heat is affected only slightly with air-movement under about 65°, above which temperature air-movement displaces the point at which the curve begins to rise, e.g. at 100 ft. per min. the rise in evaporation rate begins at about 80°.

When computing the air quantity, only the sensible heat of the occupants need be taken, e.g. 300 B.T.U./hr. at 70°. The latent heat must be taken into account when calculating the humidity gain. This gain is often taken at 0·11 lbs. per occupant per hour. Taking the latent heat at 1050 B.T.U./lb. at normal room temperature, this gain is equivalent to 115 B.T.U./hour. It may, of course, be greater, due to the activity of the occupants, in which case the figure of 0·11 lbs. would be altered to suit.

The moisture is generally expressed in grains (7000 gr. = 1 lb.).

(*e*) **Heat from Lights, Motors, etc.**—This is dealt with fully on p. 43, to which reference should be made.

Fig. 250 summarizes the various heat gains for a typical system, including duct gains, gain due to electrical energy input, etc.

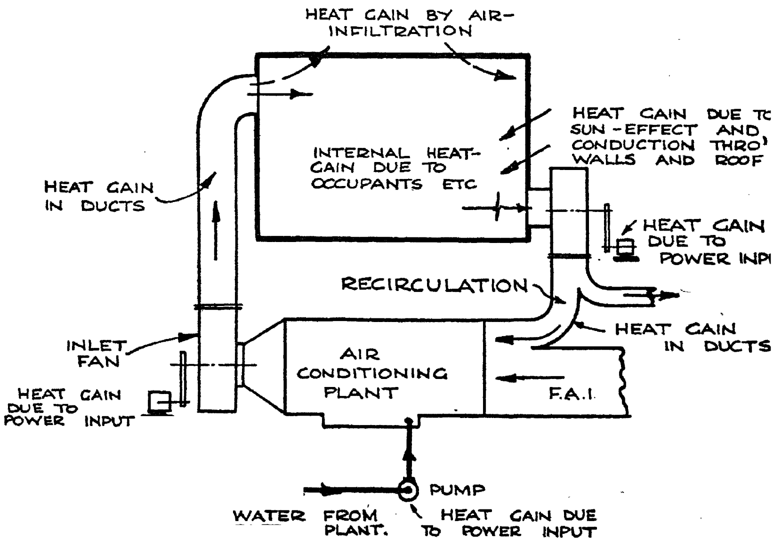


FIG. 250.—Heat Gain in Air-Conditioning.

Having established the total heat gain, the volume to be delivered to the room is found, as has already been shown, by dividing this figure by the specific heat × temperature rise. The result may be in lbs. of air or in cubic feet. The former is more convenient as the volume varies at every stage through the plant.

SOLAR COEFFICIENTS

Sun shining on a horizontal surface in June delivers approximately 265 B.T.U./hour per sq. foot of surface in Latitude 52° N. Of this heat received on the surface some is absorbed by the material if opaque, or transmitted through it if translucent. The rest is reflected from it, the relative amounts depending on the colour and nature of the surface.

Glass—For 52° N. latitude the peak sun heat transmitted through glass may be taken as follows for all orientations exposed to sun.

Vertical windows - - - -	150 B.T.U./sq. ft./hr.
Flat skylights - - - - -	190 ,, ,,
Sloping skylights - - - -	250 ,, ,,

The times at which the peaks occur will be:

Vertical Windows.

East	8 a.m. - - -	(May–July)
South-East	9 a.m. - - -	(March and Sept.)
South	Noon - - -	(,, ,,)
South-West	3 p.m. - - -	(,, ,,)
West	4 p.m. - - -	(May–July)

Skylights.

Flat	Noon - - -	(June)
Sloping	Depends on orientation	

Shading of Glass—The sun heat transmission may be reduced by means of blinds or shades, which are much more effective outside than inside. The following are the factors to be applied to the solar transmission coefficients to give the heat delivered to the room:

White or light curtains inside - - - - -	0·50
,, ,, sunblinds outside - - - -	0·25
Dark curtains inside - - - - - -	0·75

Windows giving on to pavements may gain something by upward reflection off the pavement, and the above factors should be increased by about 10 per cent.

Walls—It has been pointed out that glass transmits solar radiation direct to the interior, whereas walls do not. Further, with massive wall constructions there may be a time lag of several hours between the peak of the incident radiation and the maximum effect being felt inside the building. This 'flywheel' effect of massive construction has an important bearing on the magnitude of the peak cooling load, which cannot be determined by direct addition of its components. The usual method is to tabulate the heat gains and analyse them for various times of the day, so as to arrive at the worst case. This will be seen in the example which follows.

Since with walls the radiation and conduction effects are merged into one, a convenient method of calculation is as follows:

Wall area - - - - -	A sq. ft.
Ordinary transmission coeff. -	U B.T.U./sq. ʼ F./hr.
Actual temp. difference:	
Inside-outside air - - -	t_a °F.
Temperature increment due to sun - - - - -	t_s °F.

Then total heat gain (solar + conduction)

The value assumed for t_s can be made to allow for the proportion of the radiation reflected from the surface, and for the shift of the peak due to the 'flywheel' effect.

Table LXXIX gives values of this temperature increment for 52° N. latitude for the times, dates, orientations and constructions stated.

TABLE LXXIX

SUN HEAT TEMPERATURE INCREMENT (t_s) ON VERTICAL WALLS (IN DEG. F.)

Construction	Exposure:				
	E.	S.E.	S.	S.W.	W.
Light Frame and Sheeting - -	35	35	20	40	40
Solid Masonry, Concrete, etc.:					
4½″ thick - - - - -	20	20	10	20	20
9″ ,, - - - - -	5	5	0	5	5
13½″ and over - - - - -	—	—	Ignore	—	—

TIME OF MAXIMUM SUN EFFECT

External - -	6 a.m.	9 a.m.	Noon	3 p.m.	6 p.m.
Internal:					
Light Frame -	11 ,,	1 p.m.	1–3 p.m.	5 p.m.	6
4½″ Solid -	12–6 ,,	2–8	4–8 ,,	4–10	6–10
″	All day				All day

(The above table applies to 52° N. Lat. August.)

It should be noted that no account is taken in the above table of the effect of insulation, as this does not affect the time lag or external surface temperature, but is reflected in the value of U used in the formula already given. It will be seen that a massive wall without insulation has not the same characteristics as a light construction insulated to give the same transmission coefficient as the massive wall.

Table LXXIX is for dark-coloured materials. Though light colours will reduce the amount of absorption, it is doubtful in practice whether such materials can be assumed as staying light for a number of years.

Roofs—Sun effect on roofs is more important than on walls as it generally persists all day long.

The method of allowance for sun effect is exactly the same as for walls, and the following table gives the appropriate temperature increment which should be allowed:

TABLE LXXX

SUN HEAT TEMPERATURE INCREMENT (t_s) ON FLAT ROOFS (IN DEG. F.)

Construction	Temperature Increment (t_s)		Internal Peak at:
	Insulation on top	Uninsulated or Insulation underneath	
Sheeting - - - -	—	60	1 p.m.
4″ Concrete - - -	30	35	2 ,,
6″ ,, - - - - -	20	25	2–6 ,,
Heavier Constructions -	5	10	4–10 ,,
Sloping Roofs normal to Sun - - - -	Above × 1·6		—

(The above Table applies to 52° N. Lat. June.)

Though the maximum effect inside the room occurs at the times stated, the effect will continue to a lesser extent for some hours afterwards, and this should be considered in the calculations (e.g. p. 460 *).

A simple method of reducing heat from a roof is to spray water over it. The surface temperature is immediately lowered to that of the wet bulb, and the radiation effect is removed by the evaporation of the water. Quite apart from air-conditioning, the temperature of a top floor which is often well above outside air temperature in hot summer may be reduced by as much as 15° to 20° by this means.

Where the top floor has an attic space which mounts to 100° or over in hot weather some reduction can be effected by ventilating the space by means of a fan or by blowing the discharge of the extract fan through it.

PSYCHROMETRY

Psychrometry concerns the behaviour of mixtures of air and water vapour, and a knowledge of it is necessary in any air-conditioning calculations. The general principles were referred to on pp. 7 and 8. A complete study is outside the scope of this book and is dealt with in many text-books on thermodynamics.

The properties of a mixture of air and water vapour may be summarized as follows:

Dry Bulb Temperature (D.B.) is the temperature of the air as indicated by an ordinary thermometer.

Wet Bulb Temperature (W.B.) is a temperature related to the humidity of the air and to the dry bulb temperature. If a thermometer has its bulb kept

* "Total Fabric Gains."

wet, as by a wick surrounding it, the evaporation of the water will take heat from the mercury which will consequently contract and indicate a lower temperature than in its dry-bulb condition. The rate of evaporation from the wetted bulb depends on the humidity of the air, i.e. very dry surrounding air will cause more rapid evaporation than moister air at the same dry-bulb temperature. The difference between dry- and wet-bulb temperatures can thus be used as a measure of the humidity. This difference is known as the *Wet-Bulb Depression.*

The rate of evaporation, and consequently the depression, depends also on whether the wetted bulb is exposed to still or moving air. In the latter case the film of moisture-laden air round the bulb is removed more rapidly than in the former, so that the depression is greater. Some systems of psychrometry adopt one condition as a standard, some the other. It is immaterial which is used provided the instruments are suitably calibrated, so that a given pair of dry- and wet-bulb readings can be accurately converted into a measure of the actual humidity.

Sling Wet Bulb is the temperature indicated by a sling or whirling psychrometer (Fig. 251), consisting of two thermometers mounted in a frame which can be whirled by hand. One thermometer has a wetted sleeve dipped over the bulb. Assmann has found that above an air-speed of 5 m.p.h. and up to at least 90 m.p.h. the depression is independent of the air-speed.

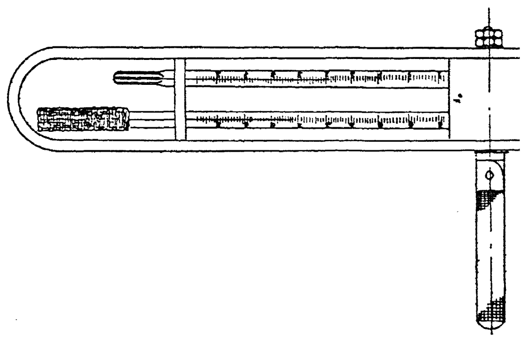


FIG. 251.—Sling Psychrometer (Negretti & Zambra).

Another type of psychrometer is the Assmann type in which the air is drawn, by means of a small fan, over thermometers mounted in a tube. In this instrument the wet bulb is kept moist by means of a wick in a reservoir.

Screen Wet Bulb is the temperature indicated by a wet-bulb thermometer in stationary air, and is so called because the standard instrument is housed in a louvred box called a Stevenson Screen.

Another approach to the conception of wet-bulb temperature is to define it as the *Temperature of Adiabatic Saturation.** This is the temperature which would be attained by moist air in intimate contact with a water surface, and in equilibrium conditions, assuming that no heat is gained or lost to an external source in the process of attaining equilibrium.

This process is a close approximation to that taking place in the wetted

* The theory of this method is due to W. H. Carrier.

wick of a wet-bulb thermometer, and at ordinary atmospheric temperatures and humidities the Sling Wet Bulb and Adiabatic Saturation Temperature differ very little, which has led to the latter being adopted in many psychrometric systems in place of the true wet-bulb temperature.

This substitution has the great advantage that conditions of equal Adiabatic Saturation correspond to equal Total Heat, i.e. the wet-bulb temperature becomes a criterion of the Total Heat regardless of changes of dry-bulb temperature.

Dew-point Temperature is the temperature to which a mixture of air and water vapour must be reduced to produce condensation of the vapour. At the dew-point the air is said to be *saturated*, and in this condition the dry-bulb and wet-bulb temperatures both coincide with the dewpoint.

Absolute Humidity is the weight of water vapour contained in unit weight or volume of the mixture. This is expressed in grains per lb. or per cubic foot, and is generally given per lb. or cubic foot of *dry air*, and not per unit of the mixture. The absolute humidity per unit *volume* depends only on the temperature, as in the case of steam.

Vapour Pressure is the partial pressure exerted by the water vapour, and is usually expressed in millibars or inches of mercury.

It follows from the above that the vapour pressure at any condition is dependent only upon the absolute humidity, and is independent of the pressure of the mixture.

Relative Humidity (R.H.) is the ratio of actual vapour pressure to the vapour pressure exerted when the air is saturated at the same temperature, expressed as a percentage. At the saturation temperature the relative humidity is 100 per cent., and dry bulb, wet bulb and dew-point are the same.

At ordinary humidities and at temperatures up to 100° F. the vapour pressure is substantially proportional to the absolute humidity, from which it follows that the relative humidity, as defined above, is approximately equal to the ratio of the absolute humidity at the given condition to that of saturation at the same D.B. temperature. This ratio may be called the *Percentage Humidity*, and is a convenient figure for rough calculation.

Total Heat or enthalpy is the sum of the sensible and latent heats of the air and of the moisture contained in it at any given condition, reckoned from an arbitrary datum, which in this book is taken at 32° F. in accordance with the method adopted in the *I.H.V.E. Guide* and in the *Callendar Steam Tables*.

Volume—The relationship between pressure, volume and temperature have been referred to on p. 14.

Specific volume is the volume of the mixture in cubic feet per lb. of dry air. Total heats and weights of moisture are referred to weights of dry air. Air containing water vapour is less dense than dry air, hence the volume occupied is greater per lb.

Table LXXXI gives the data regarding volume and weight of saturated vapour per lb. dry air, and is consistent with Fig. 252.

TABLE LXXXI

VOLUME OF DRY AIR, AND WATER VAPOUR REQUIRED TO SATURATE IT, PER LB. DRY AIR, AT VARIOUS TEMPERATURES

Temp. °F.	Volume in Cub. Ft. per Lb. of Dry Air	Wt. of Saturated Vapour per Lb. of Dry Air: Grains	Temp. °F.	Volume in Cub. Ft. per Lb. of Dry Air	Wt. of Saturated Vapour per Lb. of Dry Air: Grains	Temp. °F.	Volume in Cub. Ft. per Lb. of Dry Air	Wt. of Saturated Vapour per Lb. of Dry Air: Grains
30	12·52	24·4	55	13·16	65·2	75	13·67	133·0
32	12·58	26·8	56	13·19	67·7	76	13·70	137·7
34	12·63	29·0	57	13·21	70·2	77	13·73	142·4
36	12·68	31·4	58	13·24	72·9	78	13·75	147·5
38	12·73	34·1	59	13·27	75·5	79	13·78	152·5
40	12·78	36·9	60	13·29	78·4	80	13·80	157·8
41	12·81	38·3	61	13·32	81·2	81	13·83	163·2
42	12·83	39·9	62	13·34	84·2	82	13·85	168·8
43	12·86	41·4	63	13·37	87·3	83	13·88	174·6
44	12·88	43·1	64	13·39	90·5	84	13·90	180·5
45	12·91	44·8	65	13·42	93·8	85	13·93	186·7
46	12·93	46·5	66	13·44	97·1	86	13·96	193·0
47	12·96	48·3	67	13·47	100·7	87	13·98	199·6
48	12·98	50·2	68	13·50	104·2	88	14·01	206
49	13·01	52·1	69	13·52	108·0	89	14·03	213
50	13·04	54·1	70	13·55	111·9	90	14·06	220
51	13·06	56·2	71	13·57	115·9	92	14·11	235
52	13·09	58·3	72	13·60	119·9	94	14·16	251
53	13·11	60·6	73	13·62	124·2	96	14·21	268
54	13·13	62·9	74	13·65	128·6	98	14·26	286
						100	14·31	305

From *I.H.V.E. Guide*. At atmospheric pressure 1000 mb.

Barometric Pressure—The standard atmospheric pressure adopted for meteorological purposes in this country is 29·5306″ Hg., which is equivalent to 1000 millibars (mb.), and this is the standard used in the *I.H.V.E. Guide*. Another standard pressure often used is 29·921″ Hg. (1013·2 mb.), which is the basis of the *Callendar Steam Tables*.

For different barometric pressures the volume per lb. of mixture and of dry air will alter, and tables may be constructed giving psychrometric data for other pressures if required. For normal applications deviations in barometric pressure are ignored, but might have to be considered for air-conditioning plants installed at high altitude above sea-level. Thus the densities of air at various altitudes are:—

Sea level	1·000
2000 ft. -	·926
4000 ft. -	·858
6000 ft. -	·795

Psychrometric Chart—The relationships between the various properties of a mixture of air and water vapour may be expressed in the form of tables,

such as those published by the I.H.V.E.* Another convenient method is by means of a chart such as that given in Fig. 252, which has been prepared from the I.H.V.E. data and is obtainable to a larger scale and covering the full temperature range up to 136°.†

The method of using the chart is indicated by the diagram, Fig. 252 (*a*). The example given is for 60° D.B. 60 per cent. R.H. The chart is constructed as follows:

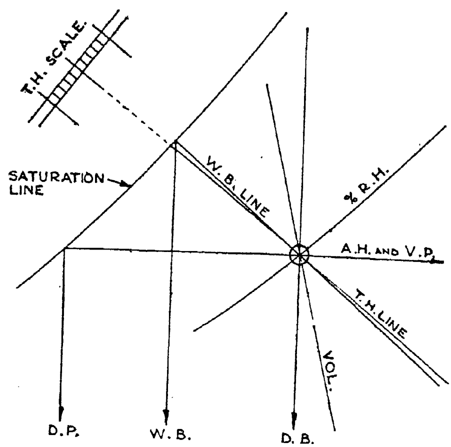


FIG. 252 (*a*).—Example of the use of the Psychrometric Chart.

Point marked is 60° D.B., 60% R.H. Other data read from the chart is:

W.B.	=52·4°.
D.P.	=46·1°.
Abs. Hum.	=46·7 gr./lb.
T.H.	=14·0 B.T.U./lb.
V.P.	=10·61 mb.
Vol.	=13·43 cub. ft./lb.

Dry Bulb—Evenly divided horizontal scale with divisions of 1° F. Read vertically to the required point.

Absolute Humidity—Evenly divided vertical scale reading in grains per lb. of dry air with divisions of 1 gr./lb. Read horizontally.

Vapour Pressure—Vertical scale corresponding to Absolute Humidity and with divisions of one mb. Read horizontally.

Relative Humidity—Indicated by the curves for each 10 per cent. Intermediate percentages can be obtained by interpolation.

Wet Bulb—Sloping continuous lines for each 1° W.B. Intermediate temperatures can be obtained by reading up from the D.B. scale onto the saturation line and then parallel to the nearest W.B. line: or by interpolation.

Total Heat—Sloping full lines for each 1 B.T.U. per lb. of dry air. Intermediate values obtained by drawing a line through the required point parallel to the nearest T.H. line, and reading onto the inclined T.H. scale.

Dewpoint—This depends only on Absolute Humidity. Hence, through the required point draw a horizontal line to cut the saturation line. From the intersection read vertically onto the D.B. scale.

Volume—Sloping broken lines of equal volume are given for each 0·2 cub. ft. per lb. of dry air. Intermediate values by interpolation.

The chart and table LXXXI have been used in the example which follows (p. 458).

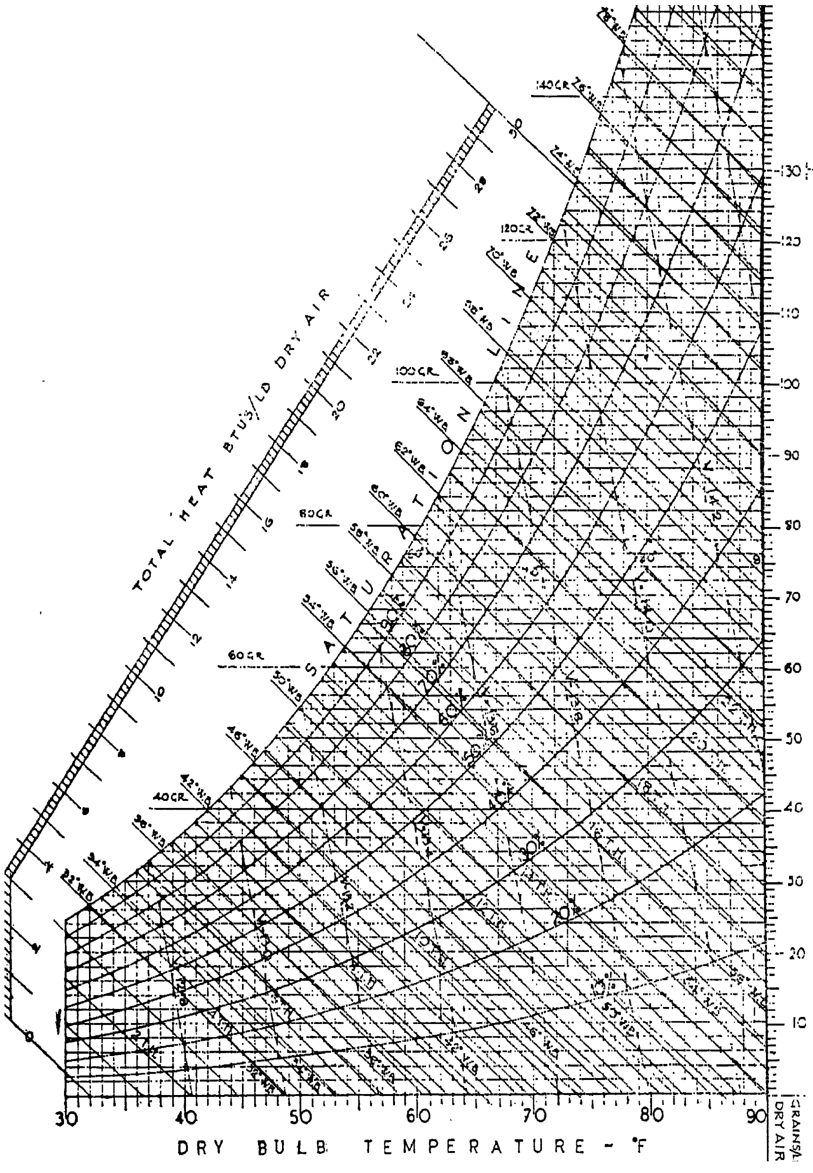
* Obtainable from the Institution of Heating and Ventilating Engineers, price 3/- post free.

† This larger chart is printed in three colours, size 40″ × 30″, and is obtainable from Messrs. Edward Arnold & Co., 41 Maddox Street, London, W. 1. Price 4/-, or post free in tube 4s. 9d.

FIG. 252.—Psychrometric Chart.

Based on Institution of Heating and Ventilating Engineers' *Tables of Hygrometric Data for Air.*

Atmospheric Pressure: 1000 MB. =29·5306″ Hg.



Ton of Refrigeration—The cooling load is often expressed in terms of the 'Ton of Refrigeration', which is the heat required to melt one U.S. ton of ice (i.e. 2000 lbs.) at 32° F. per twenty-four hours

$$=\frac{2000 \times 144}{24} = 12{,}000 \text{ B.T.U./hour.}$$

EXAMPLE OF AIR-CONDITIONING CALCULATIONS.

(a) DESIGN OF PLANT FOR SUMMER COOLING AND DEHUMIDIFICATION

Having arrived at the maximum hourly heat gain for the rooms to be conditioned, it is necessary to calculate the conditions to be maintained in the plant, and the capacity of the refrigeration unit, pump, etc.

Method—The steps in the process may be summarized as follows:

(*a*) Calculate heat gains in room.

(*b*) Hence total air quantity for permissible temperature rise.

(*c*) Check the air per occupant and the hourly air-change.

(*d*) Find amount of moisture per lb. of air, introduced by occupants, and by infiltration.

(*e*) Desired humidity at outlet from room is known, hence deduct (*d*) to find humidity at room inlet.

(*f*) Allow for heat gain through inlet duct and from inlet fan. Hence find Plant Exit Condition, humidity being as (*e*).

(*g*) This humidity corresponds to a certain dewpoint which is that required to be maintained at the plant outlet.

(*h*) Apportion air quantities between inlet, extract, re-circulation and fresh air intake to give the required minimum amount of fresh air, and to maintain a slight excess inlet in the room.

(*j*) Hence find the condition of the mixture of fresh and re-circ. air entering the plant, allowing for extract duct gain..

(*k*) Cooling load of washer is the difference of the total heats of (*g*) and (*j*). Hence calculate Washer Data.

(*l*) Size re-heater.

(*m*) Find refrigeration load, allowing for heat gain from pump, washer casing, piping, etc. Hence cooled water temperature leaving refrigerating plant.

(*n*) Check fan volume for condition as finally determined.

Basis of Design

Building - - - - -	Public Hall.
Seating capacity - - - -	500
Lighting - - - - -	10 K.W.
Cube - - - - - -	109,000 cu. ft.
Internal conditions required, summer	
External conditions assumed, summer	
Fresh air per occupant (minimum) -	600 cu. ft. per hour.

The building is screened by other buildings on three sides; west side and roof are exposed.

Other details as given in the course of the calculations (and see Fig.

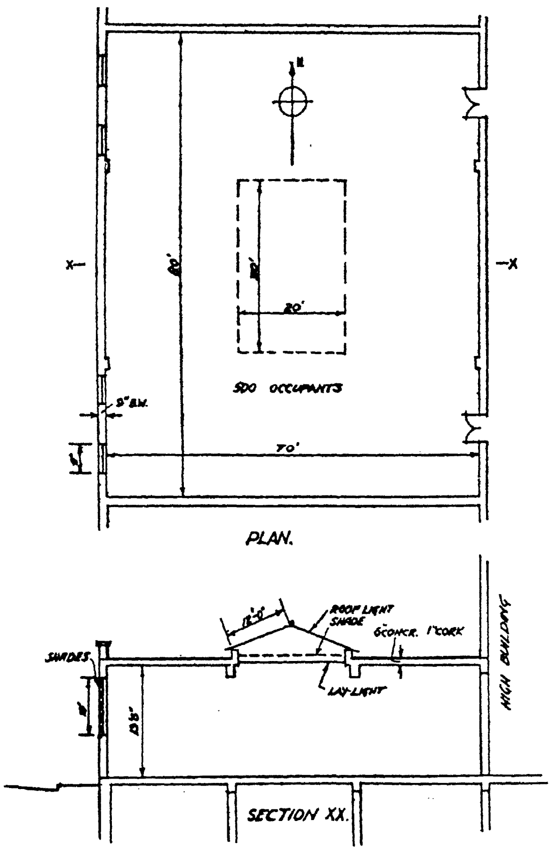


FIG. 253.

Heat Gains—

	Area sq. ft.	Transmn. Coeff. Inwards	B.T.U. per 1° F.	Temp. Diff. °F.	Total B.T.U. Conduction	Sun Effect	Total B.T.U./hr. including Sun Effect	Time of Incidence (Max.) see Tables LXXIX and LXXX
(*a*) West Windows (Single) Light shades externally. Windows look on to street	200	1·00 less 25%* = 0·75	150	10°	1500	150 B.T.U./sq. ft. × 0·25 (for shades) × 1·1 (for pavement reflection) = 42 B.T.U./sq. ft. × 200 = 8400	9,900	4 p.m.
(*b*) Roof Glass. Lantern and laylight with light internal blind	600 (plan area of double light). 360 (sloping area of half light)	0·55 less 14%* = 0·47	280	10°	2800	250 × 0·5 (for blind) = 125 B.T.U./sq. ft. × 360 = 45,000	47,800	4 p.m.
(*c*) Roof. 6″ Concrete, 1″ Cork under, Plastered inside, Waterproof Finish outside	5,000	0·20 less 7%* = 0·186	930	10° + 25° increment for sun effect = 35°			32,550	2–6 p.m.
(*d*) West Wall. 9″ Brick Plastered inside	1,360	0·43 less 12%* = 0·38	518	10° + 5° increment for sun effect = 15°			7,750	All day
(*e*) Floor to conditioned rooms under	—	—	—	—			Nil	—
(*f*) Electric Lighting	10 kw. × 3415 - - - - - - - - - -						34,150	Assume after 8 p.m.
(Total Fabric Gain)	The above do not all occur at the same time. Thus, at 4 p.m. the gain will be made up of items: (*a*) - - -						9,900	
	(*b*) - - -						47,800	
	(*c*) - - -						32,550	
	(*d*) - - -						7,750	
							98,000	
	At about 9 p.m. the gain will be items: (*d*) - - -						7,750	
	(*f*) - - -						34,150	
	Say half of (*c*) which, though past the peak, will continue to radiate heat to the room to a lesser extent - - - -						16,100	
							58,000	
	Take worst case: - - - - - - - -						98,000	
(*g*) Infiltration	Assume ½ change/hr. = $\frac{109,000}{2}$ × 0·19 × 10° =						10,350	
(*h*) Occupants	500 × 300 (sensible heat only, seated). *Note:* If the occupants were not seated, but were, for instance, dancing, the number would be less. Assume that the seated condition gives the greatest heat-gain - - - - -						150,000	
	Total simultaneous heat-gain - - - - -						**258,350 B.T.U./hr.**	

* Reduction for differing surface condition, see p. 447.

Air Quantity—

Permissible temperature rise 13° between room inlet and outlet.

$$\text{Then air required} = \frac{\text{heat gain in B.T.U./hr.}}{\text{specific heat/lb.} \times \text{temp. rise}}$$

$$= \frac{258{,}350}{\cdot 24 \times 13^\circ} \qquad = 88{,}000 \text{ lbs. of air per hour.}$$

Air Change—

Inlet temp. $=62^\circ$ F.
Spec. vol. of air at $62^\circ = 13\cdot 5$ approx. (see Fig. 252).

On given cube this represents approx. 11 changes/hour.
For given occupancy, represents 2370 c.f.h./occupant.

Moisture Gains—

Grains/hour.

(*a*) *Due to occupants.*

Moisture given off by breathing, etc., taken as 0·11 lb./occupant/hour = 770 grains/occupant/hour (1 lb. = 7000 grains).

Note: Latent heat at 75° F. = 1051 B.T.U./lb. Therefore 0·11 lbs. corresponds to 115 B.T.U./hr. (cf. p. 42).

770×500 (occupants) = 385,000

(*b*) *Due to infiltration.*

Outside condition 85° D.B. 68° W.B.:	75·5 gr./lb.	
Inside condition 75° D.B. 50% R.H.:	65·5	,,
Gain	10·0	,,

$\frac{1}{4}$ change/hour on 109,000 cu. ft. corresponds to 3930 lbs./hr. at room conditions.

$3930 \times 10\cdot 0 =$

$\therefore$ Gain per lb. of air $= \frac{424{,}300}{88{,}000}$ $= 4\cdot 82$ **grains per lb.**

Room Inlet Condition—

The permissible room conditions may be assumed to be the same as those at the extract gratings.

$\therefore$ Inlet condition $= 65\cdot 5 - 4\cdot 82$ $= 60\cdot 68$ gr./lb.
Inlet D.B. as previously determined $= 62^\circ$ F.

The conditions and quantities, as they are calculated, may be marked on a diagram, which, when completed, will be as shown in Fig. 254.

In marking the conditions on the diagram a convenient notation is to indicate the dry bulb, wet bulb, total heat and absolute humidity in the four corners of a square, thus:

D.B.	W.B.
T.H.	A.H.

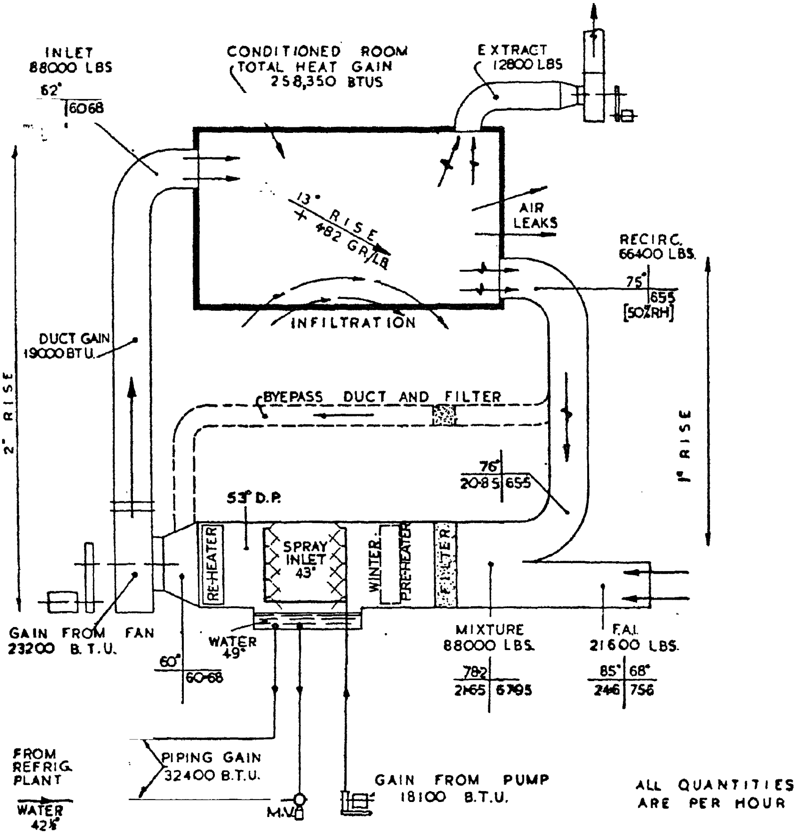


FIG. 254.—Diagram for Air-Conditioning Calculation.

Plant Exit Condition—

This will differ from the Room Inlet by the amount of heat picked up between the two points, i.e. through the walls of the inlet duct and from the fan.

(*a*) Assume 4400 sq. feet of lagged duct in air at 80°. *U* for duct $=0{\cdot}24$.
Air inside at 62° approx.
$\therefore$ Temp. difference $=18°$.
$\therefore$ Heat gain $=4400 \times 24 \times 18° =$ 19,000 B.T.U./hr.

(*b*) Inlet fan, $1\frac{1}{2}''$ W.G. 50% efficiency.
$1''$ W.G. $=68$ ft. air column.
Air quantity $=88{,}000$ lbs./hour.

$$\text{B.H.P. absorbed} = \frac{88{,}000 \times 68 \times 1\frac{1}{2}}{60 \times 33{,}000 \times 0{\cdot}50} = 9{\cdot}1 \text{ B.H.P.}$$

$9{\cdot}1 \times 2544 =$ 23,200 B.T.U./hr.

42,200 „

$$\therefore \text{Temp. rise through inlet fan and duct} = \frac{42{,}200}{88{,}000 \times {\cdot}24} = 2{\cdot}0°\text{F.}$$

Therefore plant exit condition is
Moisture as at room inlet

Apparatus Dew-point—

For a washer working as a dehumidifier it may be assumed that the contact between air and spray is sufficiently intimate to ensure that none of the air passes through the washer without being dehumidified. The washer exit condition or *Apparatus Dew-point* is thus determined by finding the Dew-point corresponding to the Absolute Humidity of the Plant Exit Condition as already calculated; in this case

D.P. for 60·68 gr./lb. = 53°

Air Supply—

Total extract to be 90% of inlet, i.e.	79,200 lbs./hr.
Fresh air to plant 600 c.f.h./occupant at room condition, equivalent to	21,600 lbs./hr.
∴ Recirc. to plant 88,000 − 21,600 =	66,400 „
∴ Top extracts to remove 79,200 − 66,400 =	12,800 „

Washer Inlet Condition—

Allowing 1° extract duct gain:		
66,400 lbs.	Recirc. reaches washer at	76° D.B. 65·5 gr./lb.
	of which the Total Heat is	20·85 T.H.
21,600 lbs.	Fresh Air at outside conditions	85° D.B. 68° W.B.
	of which the Absolute Humidity is	75·5 gr./lb.
	and the Total Heat is	24·6 T.H.
This gives a mixture at the washer inlet of		78·2° D.B. 67·95 gr./lb. 21·65 T.H.

Cooling Load—

The amount of cooling which the washer has to do can be arrived at from the Total Heat and air quantity as follows:

Entering washer	21·65 T.H.	
Leaving washer	14·44 T.H.	
88,000 lbs./hr. × 7·21 =		634,500 B.T.U./hr.

Washer Data—

Outlet D.P. already found	53°
Allow differential between water outlet and air outlet	4°
∴ Water outlet temperature	49°
Allow water temperature rise	6°
∴ Cooled water inlet temperature	43°

Quantity of water for 6° rise

$$\frac{634{,}500}{6 \times 1{\cdot}0 \text{ (sp. ht.)}} =$$

Pump pressure, say, 80 ft. head

Water H.P. of pump

33,000 4·26

Pump 60% efficient,

$$\text{B.H.P. input} = \frac{4{\cdot}26}{0{\cdot}6} = 7{\cdot}1 \text{ B.H.P.}$$

Say pump motor for 10 H.P.

Absolute Humidity, washer inlet 67·95 gr./lb.

„ „ „ outlet 60·68 „

88,000 lbs. × 7·27

= 640,000 gr./hr.

= 9 galls. per hour approx.

This quantity of water will be deposited in the washer tank and will run to drain.

Reheat Load—

The reheater has to raise the temperature of the air leaving the washer (at 53°) to 60°, and, therefore, has to supply heat as follows:

Leaving heater	16·18 T.H.		
Entering „	14·44 „		
88,000 lbs./hr. ×	1·74	=	153,000 B.T.U./hr.

Alternatively, this reheating could be affected by bye-passing a proportion of the air as indicated on the diagram.

Refrigeration Load—

The total amount of heat to be extracted from the cooling water by the refrigeration plant is arrived at as follows:

	B.T.U./hr.
(*a*) Cooling performed by washer =	634,500
(*b*) Pump heat input 7·1 × 2544 =	18,100
(*c*) Heat gains to piping, washer casing, etc. Take 5% of (*a*) and (*b*), say	32,400
	685,000
This corresponds to $\frac{685{,}000}{12{,}000}$ =	57·1 tons Refrig.
Add margin to cover losses from refrigerant piping, etc.	
10% of 57·1	5·7 „ „
Compressor Duty	**62·8 tons Refrig.**

Cooled Water Temperature—

Required water temperature at washer		43°
Heat input as above; pump	18,100	
Water piping	32,400	
	50,500 B.T.U./hr.	
Quantity of water =	105,750 lbs./hr.	

∴ Temp. rise = $\frac{50{,}500}{105{,}750}$ = 0·5° approx.

∴ Water Temperature leaving refrig. plant =

Fan Volume—

Air quantity 88,000 lbs./hr.
Air at 60° D.B. 60·68 gr./lb.
Sp. vol. from Fig. 252 = 13·47 cu. ft./lb.

$\overline{\quad 60 \quad}$ 19,750 c.f.m.

Cooling by Direct Expansion—Cooling by this method has already been described (see Fig. 246).

To avoid cooling all the air down to a low dew-point for dehumidification and subsequently reheating, advantage may be taken of a characteristic of coil surfaces whereby moisture can be deposited on the chilled surface, without the bulk of the air being reduced to the same temperature. This is achieved by keeping the surfaces at the lowest possible temperature consistent with absence of freezing.

If, on the other hand, sensible cooling is the main consideration, this is achieved most economically by a cooler having a surface at a temperature above the dew-point so that condensation does not occur.

In some plants it may be found desirable to adopt an arrangement using brine as a cooling medium, which acts as a reservoir to smooth out the operation of the plant and to provide a steadier load on the compressors.

The calculations for cooling by direct expansion will follow the same principle as the example already given, except that relative amounts of sensible and latent cooling will need to be determined separately, and air velocity and cooling coil surface areas and characteristics chosen to suit. Makers' data are required on this point.

This system is not controlled by a fixed dew-point, but by thermostat and humidostats in the return air duct as discussed later. This is in effect the same as putting the instruments in the room, any differences due to extract-duct heat gain being offset by altering the thermostat setting.

(b) DESIGN OF PLANT FOR WINTER WARMING AND HUMIDIFYING

The operation in winter may be with partial recirculation, as in summer, or with 100 per cent. fresh air. The plant might be arranged so as to be suitable for the latter method of operation, on average spring and autumn weather, when neither cooling nor heating is required. In cold weather recirculation might be used for economy of running.

The following calculation assumes 100 per cent. fresh air at 40°, below which it is assumed that recirculation would be used. This is often done by an outside control operating motorized dampers to vary the proportion of fresh to recirculated air.

Basis of Design—

Fresh air 40° D.B. saturated. (7·61 T.H. 36·9 gr.)
Room inlet 63° D.B., 60% R.H. (15·55 T.H. 52·0 gr.)
Inlet ducts in warmed spaces, no loss.
Fabric losses offset by direct radiation.
Air quantity as for summer, 88,000 lbs./hr.

The heat necessary for warming and humidifying the incoming air is divided into two parts:

(*a*) Heat supplied by preheater or spray water heater in washer circulation.
(*b*) Heat supplied by main heater.

Preheater or Spray Heater—

When an air washer is acting as a humidifier, it cannot be assumed that all the air passed will be 100% saturated. In winter, only one bank of sprays is generally used, and in this case the air may be as leaving the washer at 85% R.H.

Abs. Hum. reqd. at washer outlet = 52·0 gr.	
This corresponds to 53° D.B., 85% R.H.	13·16 T.H.
Washer inlet 40° sat.	7·61 T.H.
88,000 lbs./hr. ×	5·54 = **487,500 B.T.U./hr.**

If this heat were added to dry air by the preheater it would have the effect of raising the temperature of the incoming air by $\frac{5\cdot54}{0\cdot24}=23°$. Thus, the heater outlet and washer inlet would be 40° + 23° = 63°. In the process of humidification in the washer the temperature would be reduced by the evaporation to 53° as above.

The thermostat on the washer outlet would be set at 53°, which is a *false dew-point* at which the plant is controlled.

The quantity of water evaporated is:

Washer outlet	52·0 gr.	
„ inlet	36·9 „	
88,000 lbs./hr. ×	15·1	=133,000 gr./hour.

or 1·9 galls. per hour.

This will be supplied through the washer make-up ball-valve.

Main Heater—

Leaving heater	15·55 T.H.	
Entering heater from washer	13·15 „	
88,000 lbs./hr. ×	2·40	=**211,200 B.T.U./hr.**

This load being greater than the summer reheat load (153,000 B.T.U.) required of the heater, the size of the latter will be determined by the winter condition.

Total Heat Load—

Pre-heater	487,500 B.T.U./hr.	
Main heater	211,200	„
	698,700	„

(c) SPRING AND AUTUMN OPERATION

For a fair portion of the year, when the outer air is at a temperature of about 60°, it may be delivered into the building without cooling or warming. The system already discussed would be used, introducing 100 per cent. fresh air with no recirculation. One bank of sprays in the washer would probably be in use, giving a relative humidity of about 60 per cent.

METHODS OF WARMING AND EFFECT OF AIR-CONDITIONING

When the question of air-conditioning for a building is being considered, the problem of heating in winter has to be considered with it.

The system is then equivalent to a straight ventilation or plenum system, according to whether direct radiation is provided for taking care of the fabric losses, or not. These systems were discussed in the previous chapter.

If the space under consideration is a hall or room entirely without heat loss, such as a theatre surrounded by warmed rooms and corridors, the case is different, as no undue heating of the air for prewarming will be necessary. Even so, a small amount of direct radiation to keep the room tempered is an advantage.

ZONING

With a single hall or room operated by one air-conditioning unit, control is effected on the plant itself.

Where a multiplicity of rooms, as in an office building, are served from one air-conditioning plant the matter of controls becomes more involved if exact regulation of temperature and humidity in each room is to be provided, because the rooms may vary relatively to one another in occupancy

and sun effect from hour to hour, and strictly it would be necessary to have a separate reheater and control for each room to be conditioned.

If, however, a few degrees variation is permissible, the simplest plan is obviously to serve such a building from one plant delivering air at constant temperature and humidity with a constant volume of air supply to each room. The heat reserve of the building seems to balance out local irregularities, giving much steadier conditions than might appear possible without local control.

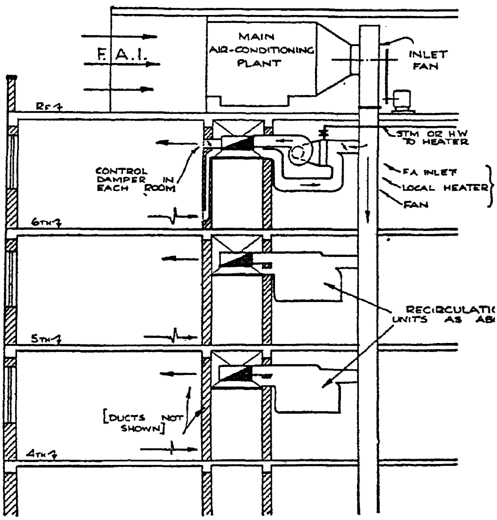


FIG. 255.—Air-Conditioning an Office Building, using Recirculation Units.

A closer approach to the ideal is obtainable if the building is divided into zones of roughly equal exposure. Thus all rooms on the east might be served from one plant, those on the south another, and so on. As the sun moves round during the day and the occupancy varies, the temperature or volume of air supplied from each plant may be adjusted automatically to suit the changing conditions.

A more complete system of control involves a thermostat in each room controlling the amount of air admitted by means of a damper in the individual supply duct. The apparatus is arranged as in Fig. 255. One plant supplies the whole building, and each floor or section of a floor has a separate recirculation unit. These consist of a small fan and heater and inlet and extract ducts to the rooms. They operate independently of the main system amd handle a relatively large volume, giving good air movement to the occupied spaces.

The suction of each recirculation fan is supplied both from its own extract duct and from the central plant. When cooling in summer the amount

of cooling air drawn from the latter is varied in response to a thermostat in one of the rooms near by. In winter the heater is similarly controlled. Having thus adjusted the temperature of the air supply for the section, it is likely that it will be nearly suitable for each room. The final regulation of the volume delivered through each outlet by the room thermostat completes the control.

As the fresh air supplied from the main plant to each recirculation unit forms only a part of the total air moved, the main system, including fan, washer, ducts, etc., may be much smaller than would otherwise be the case. For the same reason the air supply from the central apparatus in summer is at a lower temperature, probably at the dew-point. This is regulated, as previously described, by thermostatic control.

AUTOMATIC CONTROLS

Automatic controls are an essential part of any air-conditioning system. The wide variety of purposes for which such plants are required, the number of different systems which it is possible to select, and the multitude of types of control equipment available require a complete book in themselves to cover adequately.

It is possible here only to indicate the main principles involved, and some of the chief component items of apparatus commonly used. From this it will be possible to understand some of the problems involved and how they may be tackled. Other permutations and combinations of system and equipment may be built up to suit any variety of circumstances.

Three main types of plant will be considered and reference to the diagrams will be necessary along with the following descriptions.

Fig. 256—This is the type of plant envisaged in the cooling and heating calculations taken earlier.

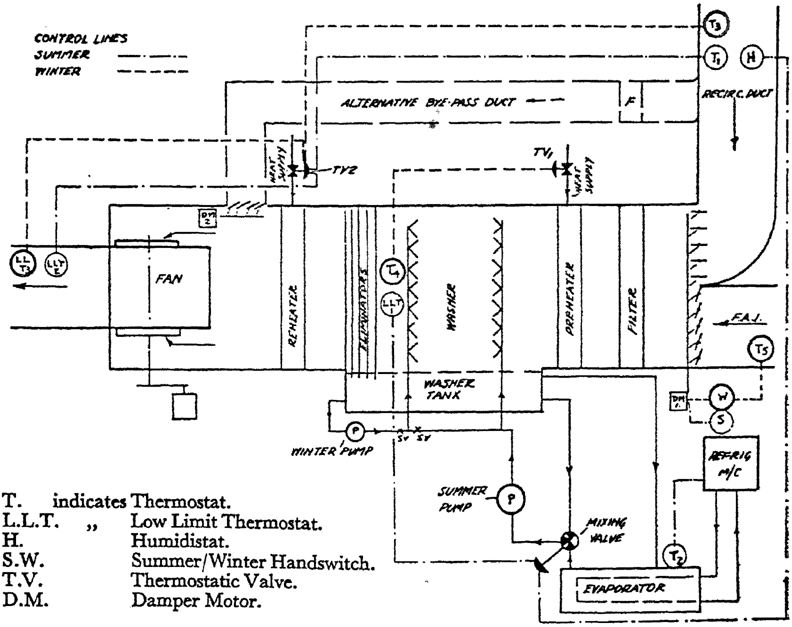
Summer Control.

Required dew-point is controlled by low limit 'stat L.L.T.$_1$, operating mixing valve in washer pump suction whereby water is drawn from the cooled water supply (from the evaporator) or from the washer tank, in the requisite proportions to give the resultant mixture at the correct temperature.

Should the Hall be empty (or partially so) the humidity gain will be less, and a higher dew-point will suffice, hence a humidistat H. is placed in the recirculation duct (equivalent to placing it in the room) so as to shut off the mixing valve against the cooled water inlet at a higher temperature.

Final temperature leaving fan is controlled by low limit 'stat L.L.T.$_2$, admitting heat to the heater through T.V.$_2$, whenever the air temperature drops below the designed condition and cutting off heat when it rises above.

Again, should the heat gain in the hall be insufficient to maintain the



T.	indicates	Thermostat.
L.L.T.	,,	Low Limit Thermostat.
H.		Humidistat.
S.W.		Summer/Winter Handswitch.
T.V.		Thermostatic Valve.
D.M.		Damper Motor.

Fig. 256.—Air-Conditioning by Spray Washer and Dehumidifier. Summer and Winter Control Diagram.

return air at the required temperature, thermostat T. will come into play and admit heat via T.V.$_2$, thus over-riding L.L.T.$_2$.

To maintain a constant cooled water outlet temperature from the evaporator, T.$_2$ stops and starts the refrigerating compressor, or in some cases may control its speed.

The proportions of recirculated to fresh air as designed will be controlled by the louvres shown, adjusted by the damper motor D.M.$_1$. This can most simply be controlled by the hand change-over switch S.W., for operation when summer cooling begins, being subsequently thrown over to the winter position. In some control lay-outs this switch is operated automatically so as to make most use of outside air when conditions permit, so reducing refrigeration running period.

If, instead of the reheater, a byepass duct is employed to bring the air to required outlet temperature, the controls operating T.V.$_2$ would instead operate damper motor D.M.$_2$, to adjust the quantity of byepass air admitted.

Winter Control.

Humidification is controlled from the thermostat T.$_4$, giving a constant dew-point, by admitting or closing the heat supply to the preheater

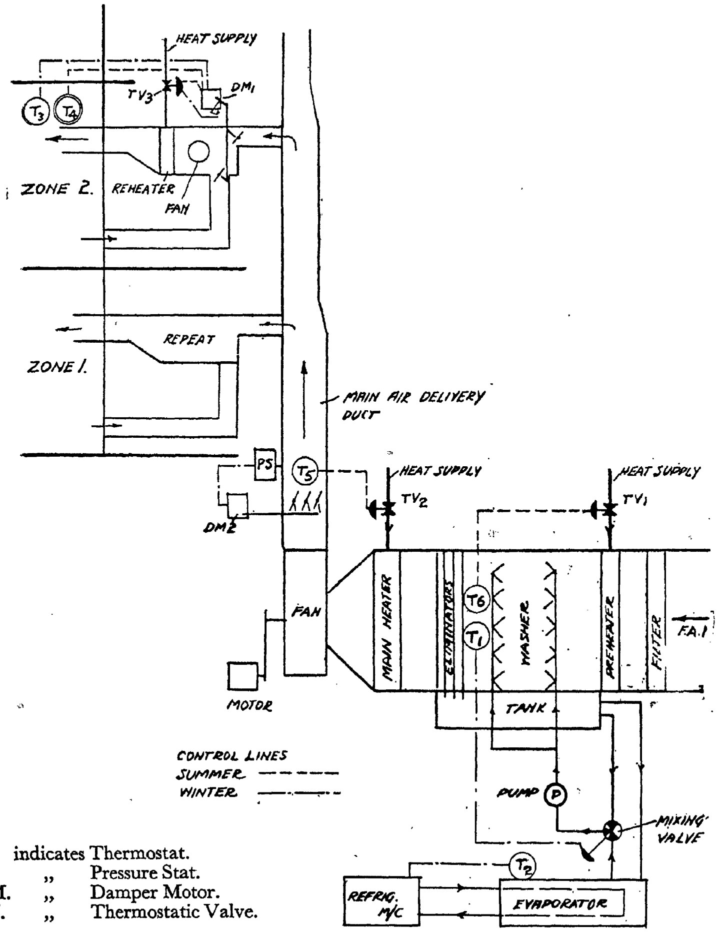
The final temperature delivered is controlled by the low limit 'stat

L.L.T.$_3$, and return air thermostat T.$_3$ acting in conjunction as explained above.

Separate thermostats are shown for winter, and are generally selected by a switch on the control panel for summer and winter operation, whichever is required being brought into use.

When the fresh air temperature falls below some predetermined point, such as 40°, economy of heat requires the fresh air to be shut down to the minimum. This is achieved by the thermostat T.$_5$ operating the damper motor D.M.$_1$ when S.W. is set to the winter position.

Fig. 257 shows a zoned system served from a central plant, as described on page 467. In this case the central plant delivers a constant outlet condition and the zones are controlled individually.



T. indicates Thermostat.
P.S. „ Pressure Stat.
D.M. „ Damper Motor.
T.V. „ Thermostatic Valve.

FIG. 257.—Air-Conditioning by Central Plant with Zone Control. Diagram of Controls.

Summer Control.

Constant dew-point outlet is controlled by $T._1$, operating this mixing valve in cooled water line as before.

The refrigerator is controlled for constant water temperature by $T._2$.

Reheating occurs at each zone according to requirements. $T._3$ in the zoned room, or in the return duct, controls $D.M._1$, so varying the proportions of cooled air taken from the plant and from the return duct. Should this adjustment continue to deliver the air too cool, $T.V._3$ is opened and heat supplied to the reheater.

When a zone reduces its air demand on the central plant, means have to be provided to reduce correspondingly the volume delivered by the main fan, or other zones will receive a surplus. This is achieved by a static pressure control device shown at P.S. On rise of pressure this causes damper motor $D.M._2$ to shut down the louvres in the discharge, so as to bring the pressure back to normal and vice versa.

Winter Control.

Humidity is controlled by constant dew-point thermostat $T._6$, operating $T.V._1$ as before.

Final temperature from the plant in a zoned system is generally set at 50 to 55°, and is controlled by 'stat $T._5$ operating valve $T.V._2$.

The reheating in each zone is controlled by the winter 'stat $T._4$ operating $D.M._1$ and $T.V._3$ as before, but in reverse. $T._4$ naturally has a different setting from $T._3$.

Changeover switches from winter to summer operation would form part of the system.

Fig. 258—Direct expansion system with separate latent and sensible coolers.

Summer Control.

Dehumidification is controlled by humidistat $H._1$ in return duct stopping and starting compressor, or in some cases controlling refrigerant admission or cooled brine admission from a central plant.

Should this reduce the air temperature too greatly, low limit 'stat $L.L.T._2$ admits heat to the heater.

Sensible cooling is controlled by $T._1$ and $L.L.T._1$, acting in conjunction as before described; and operating starting and stopping of refrigerator or valve in other cooling supply.

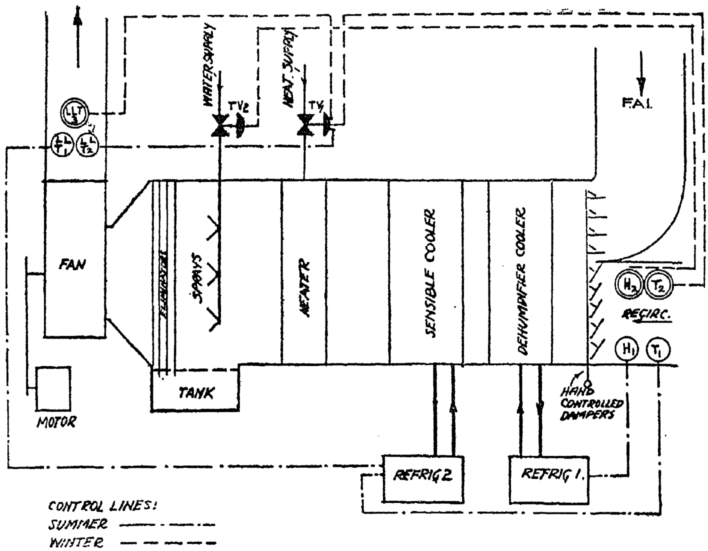
An interlock would in practice be provided to prevent the sensible cooler functioning whilst heat is being supplied to the heater.

Winter Control.

Heat supply through $T.V._1$ is controlled by $T._2$ and $L.L.T._3$, acting conjointly as previously described.

Humidity is supplied by sprays direct from a cold water line, and is controlled on or off by $T.V._2$ operated by humidistat $H._2$.

Changeover switches for summer to winter control.



T. indicates Thermostat. H. indicates Humidistat.
L.L.T. „ Low Limit Thermostat. T.V. „ Thermostatic Valve.

FIG. 258.—Air-Conditioning using Direct Expansion. Summer and Winter Controls Diagram.

AUTOMATIC CONTROL APPARATUS

There are three main classes of controls; electrically operated, pneumatically operated, and hydraulic or oil operated. The choice will be determined by preference based on experience, size of job, cost, personnel running plant, etc.

Electrical controls are best where inexperienced operators supervise the plant, but are equally suitable on any plant.

Pneumatic controls need skilled attention from time to time, and when properly adjusted give perhaps the most simple and gradual operation of all. They are often to be found on industrial plants for this reason, especially as such plants generally already include a compressed air system.

Hydraulic and oil controls have their special uses but their limitations.

The positions in which controlling elements, such as thermostatic bulbs are fixed, require careful selection. Air passing through a plant may 'layer' seriously. Near the edges of a duct or plant it may similarly be different in temperature from that near the core. Again, there is the matter of heat or cold radiation from heater batteries and cooling coils or even from cooled spray water.

To obtain as nearly average results as possible temperature sensitive or

hygroscopic elements should extend well into the middle of the stream, and not merely project a few inches through the casing. The elements should be shielded against radiation by polished shields, left open or perforated away from the radiation.

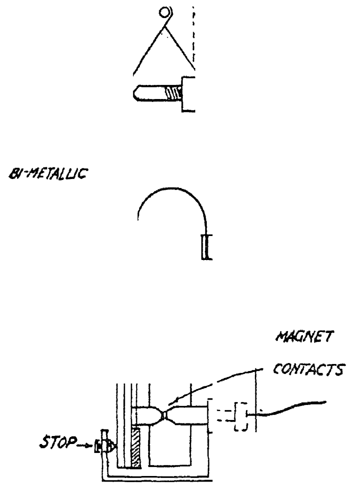


FIG. 259.—Principle of Operation of Electrical Bi-metallic On-Off Thermostat.

ELECTRICAL CONTROLS

Electrical Thermostats — These comprise a temperature sensitive element, either of volatile liquid tilting a mercury switch, or bimetallic type closing point contacts. Fig. 259, *on-off type*, 2 wire, simply opens or closes a circuit to stop or start a motor. The 3-wire type opens one circuit and closes another to start a motor or operate a valve in reverse.

These thermostats may be arranged to give a step-by-step action, opening or closing a series of contacts in sequence.

Modulating type (Fig. 260)—This works in conjunction with a modulating valve or damper motor as described later. The thermostat operates the moving arm of a potentiometer in a 'bridge' circuit.

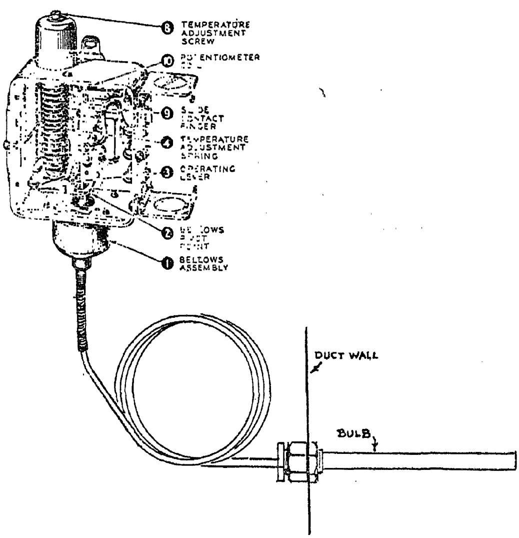


FIG. 260.—Electrical Duct Thermostat Modulating Type (Honeywell Brown).

Electrical Humidistat (Fig. 261)—The hygroscopic material is either a bundle of hairs maintained in a state of tension, or a length of specially sensitive wood or pine-cone fibre. The expansion on increase of relative humidity is arranged to open or close contacts or operate a potentiometer, as in the case of a thermostat.

Another form of humidity control is by wet and dry bulb thermostats working on a differential principle. These control for a constant difference which corresponds over small ranges with the relative humidity. The wet bulb is kept wet by a wick dipping in a reservoir of distilled water.

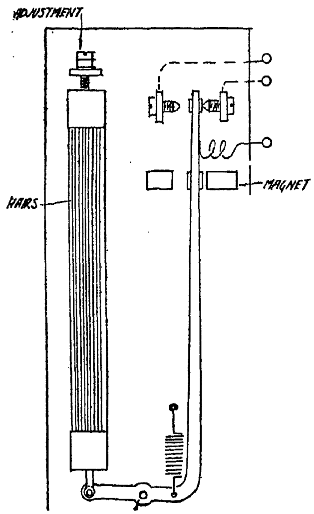


FIG. 261.—Principle of Operation of Electrical Humidistat. Two-way Type.

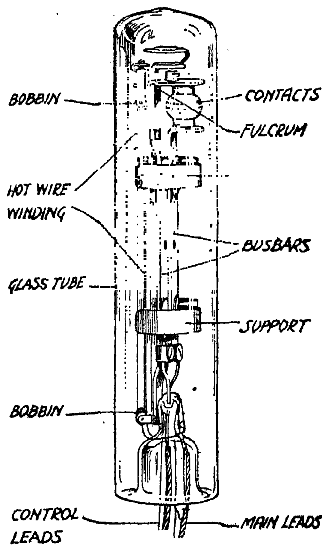


FIG. 262.—Sunvic Hot Wire Vacuum Switch (Sunvic Controls Ltd.).

Contactors—When a large machine such as a refrigeration compressor motor has to be started, it is necessary to interpose between the thermostat and the heavy current switchgear a device which will operate with the small current handled by the 'stat. This is accomplished by a contactor, which is a common piece of electrical gear. The small current serves to close the solenoid circuit, which in turn pulls in the main switch.

Fig. 262—Another device for a similar purpose is the 'Sunvic' vacuum switch, in which the small current closes a hot-wire heater circuit. The heater heats a heavy bimetallic leaf which closes the main contacts. Being in a vacuum, burning of the points is avoided.

Sequence Switches (Fig. 263)—It is sometimes necessary to operate from one thermostat a series of switches in sequence, as when two or three refrigeration compressors have to be started in turn with increases of load, or

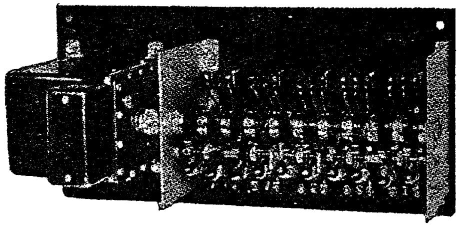


FIG. 263.—Sequence Switch.

when the air heating is accomplished by electrical heaters requiring to be switched on in steps. For this purpose a programme or sequence switch may be used as shown in the figure. Mercury switches are tilted by cams fixed on a shaft driven through gearing from a small motor controlled by the thermostat or humidistat of modulating or step-by-step type.

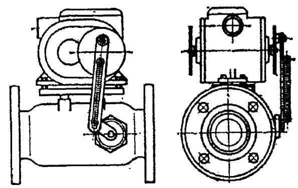


FIG. 264.—On-Off Motorized Valve for Steam (Rheostatic Co.).

EQUIPMENT CONTROLLED: ELECTRICAL

Electrically-operated Valves — These control steam or water admitted to heaters, coolers, or refrigerant admitted to coils.

Fig. 264—This shows an *on-off valve* with motor operation. A three-wire 'stat is required. The motor contains limit switches which, at the end of half a revolution, cut off and re-set for reversal.

Another type is the *stalling motor*, which opens on making of circuit and closes on opening of circuit. It thus shuts itself on current failure, a desirable feature in some cases. It requires a two-wire 'stat. Alternatively it may be arranged to operate in the opposite direction, i.e. opening when the switch closes and vice versa.

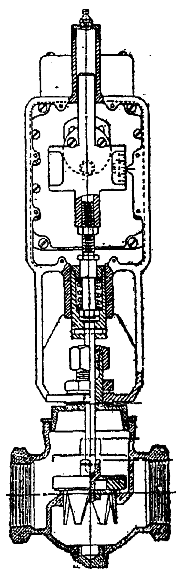


FIG. 265.—Modulating Electrically-operated Valve(Honeywell Brown).

Fig. 265—This shows a *modulating valve* operated from a modulating thermostat or humidistat. Its purpose is to adjust the valve to any intermediate position as required by the conditions. With each movement of the thermostat arm the valve 'inches' either more open or closed, where it remains until another change takes place.

The system operates at low-voltage through a transformer, the circuit being as Fig. 266.

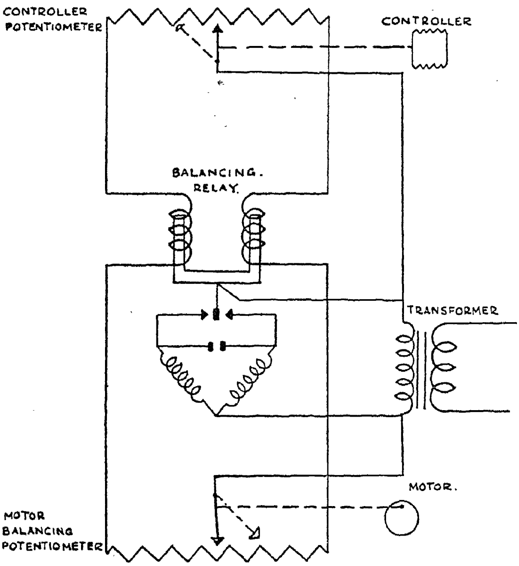


FIG. 266.—Electrical Circuit for Honeywell Brown Modulating Control.

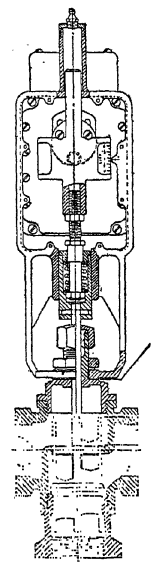


FIG. 267.—Modulating 3-Way Mixing Valve (Honeywell Brown).

Fig. 267—This shows a modulating three-way *mixing valve*, the function of which has already been discussed. This is of necessity a valve which requires gradual adjustment over its range, and the modulating system above described will normally be used with it.

Electrically-operated Dampers (Fig. 268)—For operating louvres or dampers, a *damper motor* is required. The power required to move dampers may be considerable and the motor must be chosen to suit. Sometimes the louvres will require to be sectionalized, each section being worked by one motor with linkage. Again, such damper motors may be on-off, i.e. open-shut, or they may be modulated to give settings in any intermediate position.

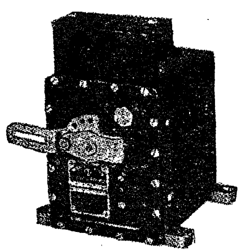


FIG. 268—Electric Stalling Motor for Damper Operation.

Control Panel—The wiring to electrical controls on a big scheme becomes very involved, and it is usual to centralize all switches, pilot lights, etc., on a panel with labelling, so that the operation can be seen at a glance.

This panel may also accommodate remote temperature and humidity indicators and recorders if required.

PNEUMATIC CONTROLS

The compressed air for these controls, if derived from a central supply, is usually taken through a reducing valve at about 15 lbs./sq. in. If the supply is provided independently a small air compressor and storage cylinder is required, automatically maintaining a constant pressure of perhaps 40–50 lbs. per sq. in. and supplying again through a reducing valve so as to give a good storage of air for sudden demands.

The piping has to be carefully run to provide adequate drainage of the water which condenses out of the air. Traps are generally fitted at low points, and other steps taken to ensure dryness. Water may completely upset operation.

Where several plants exist, a central compressor may be used piped to each unit at high pressure, each having its own storage cylinder and reducing valve.

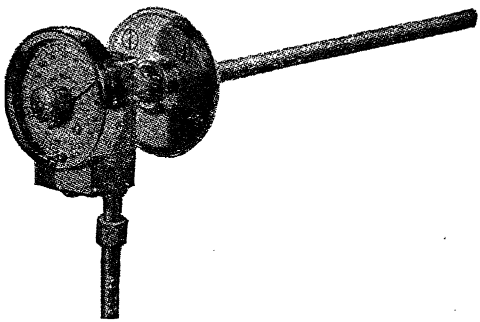


FIG. 269.—Pneumatic Duct Thermostat.

Thermostat (pneumatic) (Fig. 269)—The type shown operates on the different rate of expansion of two metals, one a rod and the other a tube surrounding it. This movement opens and closes a pilot valve to control the main unit.

Humidistat (pneumatic)—One type operates on the wet and dry bulb principle, comprising two normal thermostats connected so that when their differential changes the pilot valve is caused to open or shut. Alternatively a hygroscopic material may be used, as in electrical controls.

On-off Controls (pneumatic)—The method of operation is the same as the above, but the pilot valve is so arranged as to remain open or shut against a spring, or with a lost-motion device until beyond a certain point of movement it closes or opens suddenly.

Equipment Controlled (pneumatic) (Fig. 270)—*Pneumatic valves* operate by means of a diaphragm or copper bellows connected to the valve spindle. The air supply connects to the top of this and has a small orifice plate, or needle valve, allowing only a small quantity of air to pass. The diaphragm

top also connects to the thermostat. When the pilot valve in the latter shuts, the air pressure builds up on top of the diaphragm and depresses the valve spindle to close (or in the reverse acting type, to open) the valve. When the pilot valve opens, the pressure on the diaphragm is released and the spring around the valve spindle forces the valve up again. The air discharged from the pilot valve discharges to atmosphere. Any intermediate position of the pilot valve causes the valve to remain in an intermediate position, so giving a 'floating' control over the complete range.

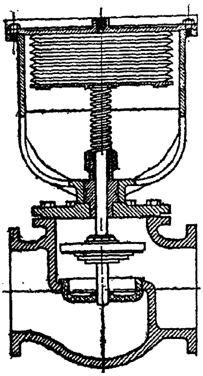


FIG. 270.—Pneumatic Valve.

A pneumatic *mixing valve* operates in the same way, as will be appreciated.

Fig. 271—Pneumatic *damper motors* work on the same principle as for valves, except that the diaphragm is much larger to give the necessary force to operate the damper.

Pneumatic Fittings—Each valve should have a control cock and pressure gauge. These are usually centralized on a board along with the thermometers and other instruments as with the electrical system, so that the operator can see the condition at a glance. The pressure gauges are connected to the diaphragm top in each case, so giving an indication as to whether the valve is open, shut, or in a mid position. A main pressure gauge on the air supply will show whether or not the compressor is functioning.

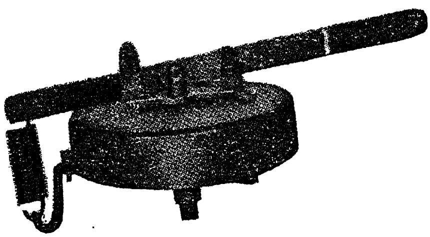


FIG. 271.—Pneumatic Damper Motor.

Electro-pneumatic—A combination of electrical and pneumatic controls may be useful where room control is required. Pneumatic room 'stats are cumbersome, and may be replaced by electrical instruments which in turn control the air supply to pneumatic valves, etc., as before. This special application will be apparent, and need not be discussed further.

HYDRAULIC CONTROLS

These operate exactly as for pneumatic, except that water at a steady pressure replaces air; the water from the pilot valves is led to drains. The

disadvantage of water is that the small ports and orifices tend to close up with deposit and require cleaning at more frequent intervals.

In another system the water is replaced with oil, pumped by a small pump, and returned from the pilot valve to a tank to be used again.

Oil Thruster (Fig. 272)—This unit operates by oil pressure derived from a small pump in the valve unit itself. The pump is driven by a motor continuously running. The thermostat bulb is of a vapour or liquid expansion type operating a small needle valve in the oil circuit. As the valve opens, pressure is applied to the top of a piston which is depressed, so closing or opening the valve. The return motion is derived from a spring. A limitation as to distance of operation is imposed by the length of capillary tube from the bulb to the valve.

Static Pressure Control—This has not been mentioned under electrical or pneumatic controls, as it is a device of special application to the zoned system, as already mentioned. In the electrical system differences of pressure adjust a modulating potentiometer controlling a damper motor for opening or shutting the dampers in the fan discharge.

In the pneumatic system differences of pressure are applied to control a pilot valve, operating a pneumatic damper motor.

Design of Valves—For control of steam, water, refrigerant, etc., valves of whatever type require to be such that the fluid controlled is metered gradually from full open to full shut, as nearly as possible in a straight line fashion. The ordinary gate or globe valve does not obey such a requirement, but on closing allows full volume to pass until within about 20 per cent. of closing, when the volume begins suddenly to fall off.

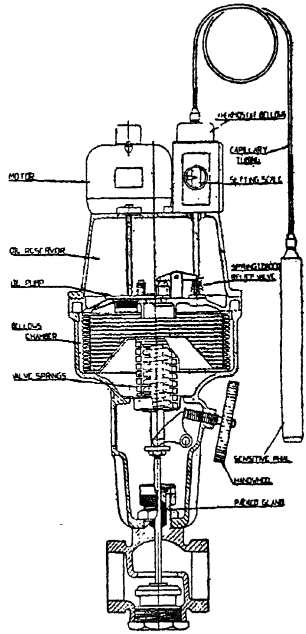


FIG. 272.—Oil Thruster Valve (British Thermostat Co., Teddington).

The desideratum is achieved by adopting valves of a Vee-port design, and such are commonly included by makers of this equipment.

As already explained, other varieties of automatic controls abound for which makers' publications and other technical literature should be consulted.

INSTRUMENTS FOR AIR-CONDITIONING

The results being achieved by an air-conditioning plant necessitate some supervisory equipment.

For a small direct expansion cooling plant for air-conditioning, say a small restaurant, perhaps nothing more than a thermometer on inlet and outlet is required.

For a large plant, or a combination of several plants in a building, more extensive instruments are advisable.

In general it may be said that wherever a thermostat exists in the system it is desirable to accompany it by a thermometer—this will be an aid to adjustment. Similarly, where a humidistat exists or a thermostat controlling humidity in some way, a wet bulb thermometer or hygrometer is desirable. As stated earlier, these are best of dial pattern distant reading type reading on the control panel.

For a multi-plant job, the essential outlet conditions may be transmitted electrically to dials or recorders in the Engineer's Office. The positioning of the bulbs requires care, as in the case of thermostats.

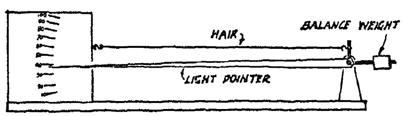


FIG. 273.—Showing Principles of Hair Hygrometer.

Various types of thermometer were referred to in Chapter I. Where relative humidity must be maintained within closed limits, a useful instrument is a portable *hygrometer*. The principle of operation is shown in Fig. 273. This instrument may equally be made to record on a paper chart turned by clockwork, and is often combined with a thermograph in one instrument, thus giving a complete record of conditions. Being portable it may be taken from place to place in a conditioned space to act as a check on plant operation.

CHAPTER XIX

Ventilation and Air-Conditioning Plant Described

Having decided on the type of air system to be employed, and made the calculations of air quantity and temperature, it is now necessary to consider in more detail the different systems of air distribution which may be utilized, and to discuss the characteristics and performance of the apparatus employed. Most of the discussion which follows applies equally to ventilating or air-conditioning systems.

AIR DISTRIBUTION

Successful air distribution, mentioned in Chapter XVII, requires that an even supply of air over the whole area be given without direct impingement on the occupants and without stagnant pockets, at the same time creating sufficient air movement to cause a feeling of freshness.

This indicates what is probably the key to the problem of successful distribution: that unduly low velocities of inlet are to be avoided, just as much as excessively high ones; and that distribution above head level not directly discharging towards the occupants will give the necessary air movement to ensure proper distribution over the whole area, and without draughts. The question of what are the best velocities and location of inlets will be discussed later.

There are four general methods of air distribution.

(1) Upward.
(2) Downward.
(3) Mixed upward and downward.
(4) Crosswise.

The choice of system will depend on:

(*a*) Whether straight ventilation or complete air-conditioning is employed.
(*b*) The size, height and type of building.
(*c*) The position of occupants and/or heat sources.
(*d*) The location of the ventilating plant, and economy of duct design.

Upward System—The air is introduced at low level and exhausted at high level as in Fig. 274, which shows a section through the 9th Church of Christ Scientist, Westminster, with mushroom inlets under the seats and riser gratings (see p. 499) in the gallery risers. The air is exhausted around

the central laylight in the roof. This system was designed so as to be reversible, i.e. it may be worked as a downward system.

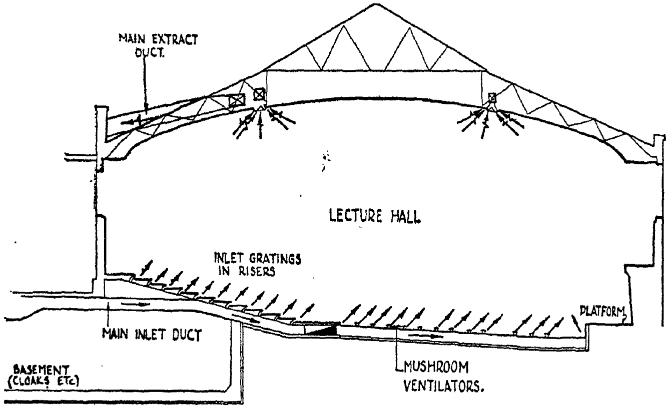


FIG. 274.—Example of Upward Ventilation.

When working upwards the air appears to be somewhat 'dead', due to the very low velocity of inlet (about 100 ft. per min.) necessary with floor outlets to prevent draughts. When working downwards more turbulence is set up in the air stream, with a greater feeling of freshness.

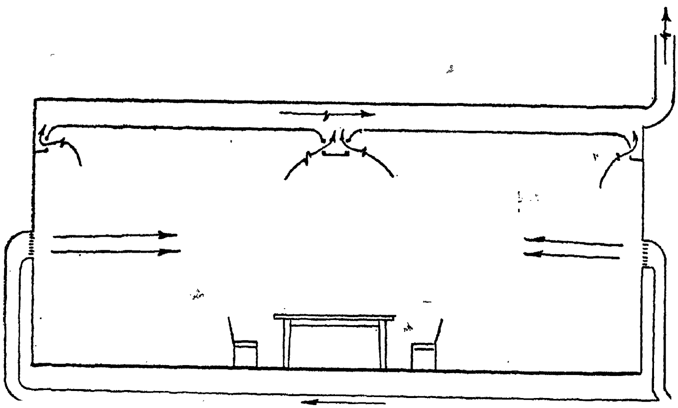


FIG. 275.—Upward System applied to Committee Room.

The upward system is not, however, confined to one with floor inlets. The inlets may equally well be in the side walls, with extract in the ceiling as before. Such a system for a Committee Room of a Town Hall is shown in Fig. 275, and another example for a theatre in Fig. 276.

The limitation of an upward system is that in a large hall it may be difficult to get the air to carry right across without its picking up heat *en route* and rising before it reaches the centre.

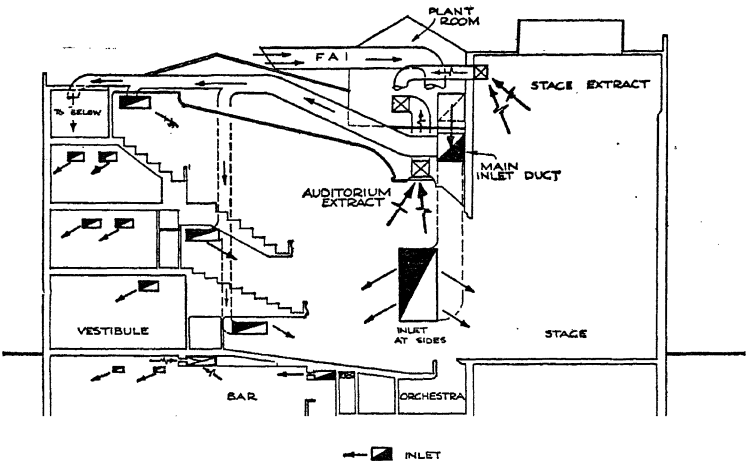


FIG. 276.—Example of Upward System in the Theatre.

The upward system is generally used with straight ventilation. When the air is cooled, as in a complete air-conditioning system, it will tend to fall too early, before diffusion, and thus cause cold draughts. The upward system also lends itself to simple extract by propeller fans in the roof in the case of a hall, factory, etc., and is thus generally the cheapest to install.

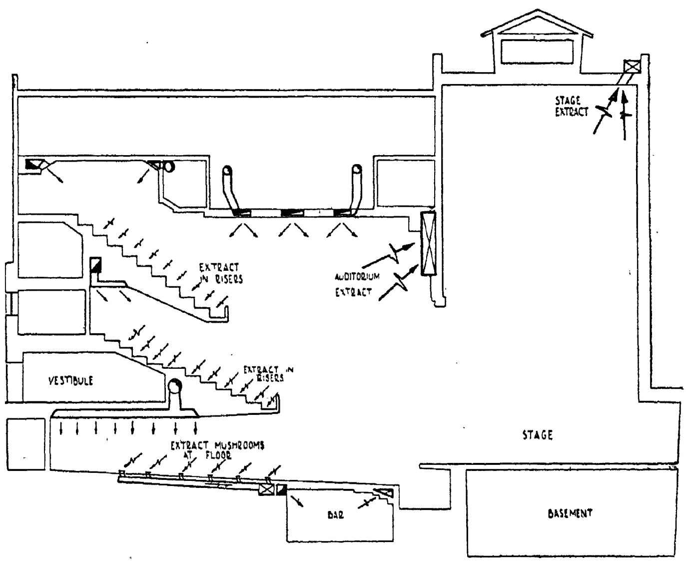


FIG. 277.—Example of a Downward System in a Theatre.
The high level exhaust labelled 'Auditorium Extract' only comes into use when the fire curtain is lowered, the normal extracts then being shut off.

Downward System—In this type the air is introduced at high level and exhausted at low level, as in Fig. 277. It is most commonly used with full air-conditioning where, due to the air being admitted cooled, it has a tendency to fall. The object of distribution in this case is so to diffuse the inlet that the incoming air shall displace the warmer outgoing air somewhat like a piston pushing downwards. Thus, the inlets shown in the figure

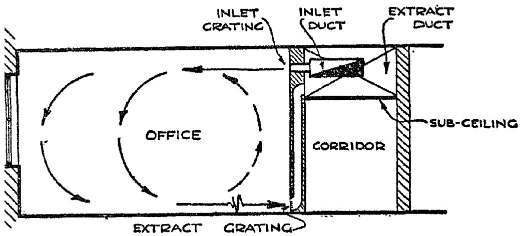


FIG. 277 (*a*).—Wall Gratings for Downward Distribution in Office Building.

as discharging downwards, in practice deliver horizontally at sufficient speed to ensure that the air completely traverses the auditorium. Turbulence is thus caused with the desirable effect already mentioned. On a smaller scale, as applied to an office building, this system appears as in Fig. 277 (*a*).

The extracts shown consist of floor mushrooms and gallery riser vents. Where the system is applied to, say, a restaurant or dance hall, where floor outlets are not possible, the extract grilles will be placed in the side wall. Another possible arrangement used by the authors in the Bank of

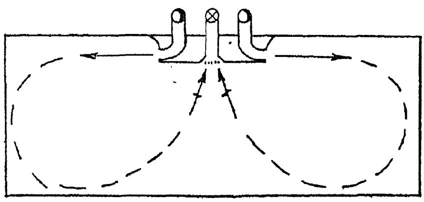


FIG. 278.—Downward-upward System, Bank of England.

England and elsewhere is a variation of this, namely, 'downward-upward', as Fig. 278.

An application of downward inlet with both downward and upward extract is shown in Fig. 279. This is usually adopted where smoking occurs and it is necessary to provide some top extract to remove the smoke. In this case the top extract is discharged to atmosphere by a separate fan and the low-level extract constitutes the recirculated air. Care must be taken to avoid placing low-level extract grilles near where people sit. If there is

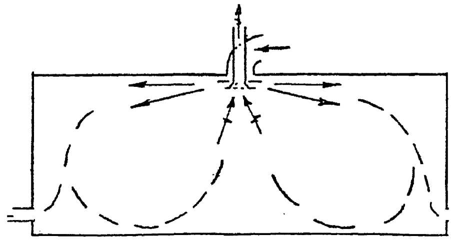


FIG. 279.—Downward System with Smoke Extract in Ceiling.

no alternative position, they must be at a very low velocity (about 150 ft. per min.) through the free area, and well spaced out, so that big volumes do not occur at any one point.

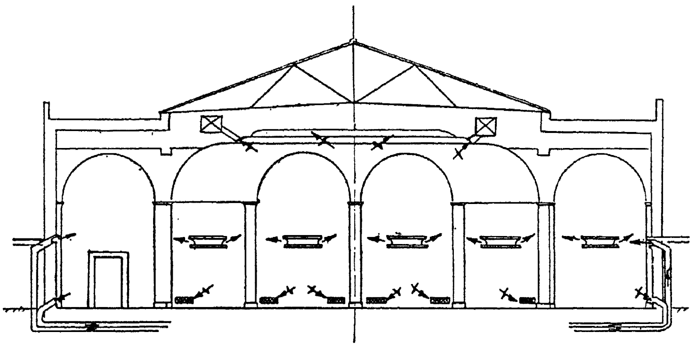


FIG. 280.—Mixed System of Distribution applied to a Restaurant.

Mixed Upward and Downward—Such a system is shown in Fig. 280. The principle will be clear from what has been described above. It is in effect an upward system giving good turbulence above head-level, with about 25 per cent. extract at low level to avoid any tendency to short circuiting from inlet straight to outlet. The remainder of the extract is exhausted normally at the ceiling.

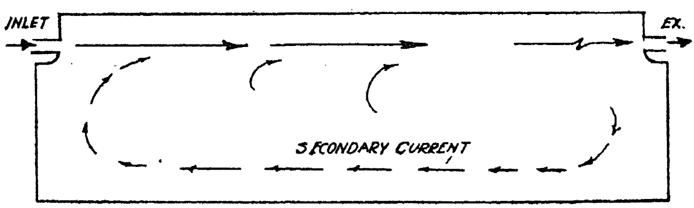


FIG. 281.—Crosswise Distribution.

Crosswise Ventilation—This has been adopted by the authors in a number of cases to meet particularly difficult cases which precluded more orthodox solutions. It has been found to work very efficiently. Air is intro-

duced near the ceiling on one side of a long low room with smooth flush ceiling, and is exhausted at the opposite end at the same level (see Fig. 281). The inlet is at a high speed and strong secondary currents are set up in the reverse direction at the lower levels, as shown. It is these secondary currents which are so important in any distribution system, causing turbulence as already mentioned.

Arrangement of Inlets—If air is delivered from a plain opening at the end of a duct as in Fig. 282, and the air is at the same temperature as the room, it will gradually diverge and its speed become less until its forward

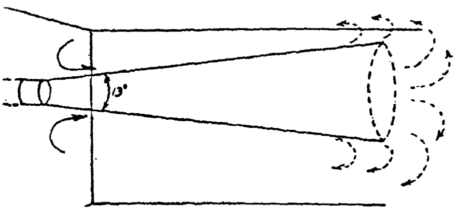


FIG. 282.—Showing Spread of Jet of Air.

movement breaks down altogether and it then becomes a series of eddies. In the process of discharge the incoming air will entrain secondary or room air to an extent depending on its speed, so that the volume of air in motion will be greater than that actually discharged.

A plain outlet may entrain about 2 parts room air to 1 of primary air at 1000 ft. per min. Thus, if

Primary air	=	500 cu.ft./min.
Entrained air	=	1000 ,, ,,
Total air	=	1500 ,, ,,

By the principle of conservation of momentum, initial momentum = final momentum; hence

$$500 \times 1000 = 1500 \times 333.$$

The velocity at the point at which 1500 cu. ft. is moving will thus be 333 ft./per min.

Similarly, as the volume has increased 3 times, and the pressure may be assumed constant, the area of the 'jet' will also have increased 3 times.

The inlet duct area was $\dfrac{500}{1000} = \frac{1}{2}$ sq. ft.

$\therefore$ The area of 'jet' where velocity is 333 ft./min. $= 1\frac{1}{2}$ sq. ft.

This spread of an air stream from a plain regular outlet is generally found to occur at an included angle of about 13°.

When the velocity falls below about 200–300 ft. per min. the entrainment of secondary air is feeble and the stream soon begins to break up, until when it has dropped to a speed of 40 ft. per min. it is considered as having reached the limit of its effective 'blow'.

The *length of blow* depends on velocity, volume and ratio of length to breadth of inlet, and on position of inlet relative to ceiling. The greater the velocity or volume the longer the throw; the nearer the shape to a circle or square the longer the blow: a long narrow grille will not throw so far as a more nearly square one. A grille near the ceiling will throw further than one halfway up the wall of a high room. The speed of inlet should be so arranged that the air has lost its velocity before reaching the opposite wall or an opposing air stream, otherwise draughts will occur.

Another matter of importance is the 'drop' of the air stream if it is entering at a temperature cooler than the room air, as in an air-conditioning system. This is a difficult matter to calculate and is best found from makers' test data. So long as the velocity is maintained above about 200 ft. per min. there will be no tendency to drop; after this the stream should have entrained sufficient air from the room to raise its temperature. Draughts will then be avoided. This explains the necessity for encouraging the entrainment of secondary air, best done by splitting up the inlet into a series of small jets of high velocity, at the same time striking a balance with the problem of length of blow.

Care in the positioning of grilles must be taken to see that the air will not impinge directly on to beams or ceiling obstructions. If such occurs the forward velocity is converted to a downward one with bad draughts.

The positioning and proportioning of inlet grilles should be such that with the length of blow to carry the air the full distance to opposite wall or to centre of hall, the spread will be just sufficient for the streams to merge into one at the limit of their travel. Thus, blowing across a narrow room, a greater number of inlets will be required than for a wider room of the same cube. Length of blow for plain nozzles and for special grilles is discussed further later.

The speed of inlet air may be limited by noise. Noise is caused by the cutting action of the air on the sharp edges of the grille. If the grille were perfectly streamlined and smooth there would be no appreciable noise even at relatively high speeds. With commercial grilles the following may be used as a guide to permissible velocities:

	Inlet Velocity, ft. per min.
Extreme silence	300 to 400′/min.
Living spaces, quiet concert halls, etc.	400 to 600′/min.
Restaurants, committee rooms, council chambers, private offices	600 to 800′/min.
General offices, cinemas, theatres	1000 to 1500′/min.
Stores, factories, exhibition halls	1500 to 2000′/min.

The grille area required is then determined thus:

$$\text{free area grille (sq. ft.)} = \frac{\text{volume cu. ft. per min.}}{\text{velocity ft. per min.}}$$

Inlet Grilles and Gratings—

Plain Lattice Stamped Steel (Fig. 283)—This is the cheapest and commonest. It permits of no control of direction, in fact often giving a quite

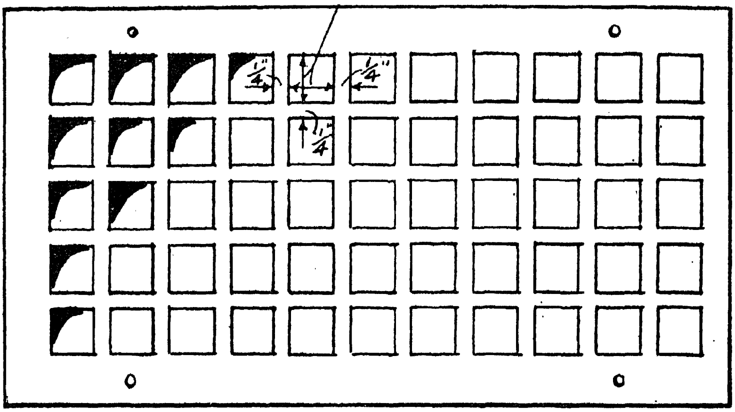


FIG. 283.—Standard Stamped Steel Grating. Free Area = 56% of Face Area.

unexpected effect, due to the angle of approach of the air from the duct, as in Fig. 284. In this case the bottom half of the grille is acting as an inlet for the entrained air from the room, and there is an unduly high air speed in the other half.

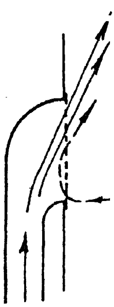
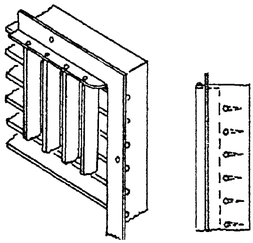


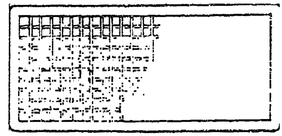
FIG. 284.—Plain Wall Grating showing unsatisfactory delivery.

With speeds much over 500 ft. per min. this grille may be noisy. It is satisfactory for small rooms with an inlet speed of about 300 ft. per minute, and for extracts at speeds of about 500 ft. per minute.

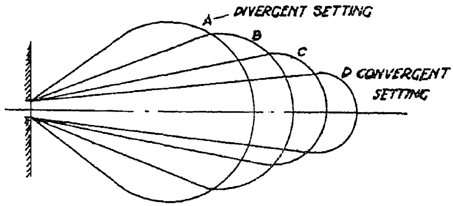
Vane Type Grilles (Fig. 285)—The figure shows a typical form of this type. Horizontal and vertical vanes of streamline section divide the air up into narrow streams. By adjustment of the vanes the air may be deflected up or down or diverging, straight, or converging. Similarly any combination of direction may be achieved, such as sideways and downwards, divergent upwards, etc. The figure also gives some characteristic curves for this type of grille. Deflectors have little influence on air direction at speeds below about 400 ft. per minute.



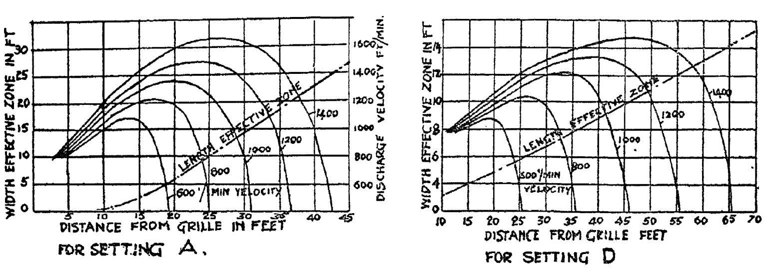
(*a*) Detail.



(*b*) Front View.



(*c*) Effect of Different Vane Settings.

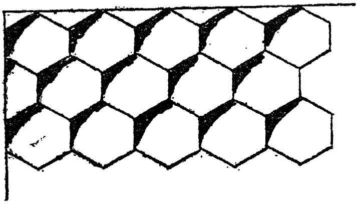


(*d*) Effective Zone and Throw.

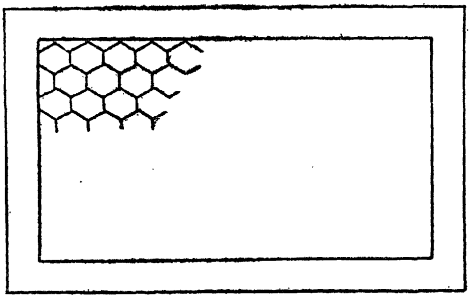
FIG. 285.—The 'Deflecto' Vane Type Grille (U.S. Air Cond. Corpn.).

'90' Grille (Fig. 286)—This type was introduced into this country from Canada some years ago, and has been used successfully by the authors on a number of installations. It consists of a series of metal pressings, fixed together in frame so as to form a honeycomb pattern. The hexagonal tubes are about $\frac{3}{8}''$ across and about $1''$ deep, giving a vane depth of about three times the width of stream and consequently ensuring directional control, again of course at air speeds of about 600 ft. per minute or over.

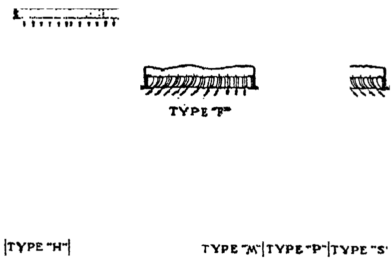
The pressings are made so that a series of types can be built up as required, as in Fig. 286 (*c*). They are not adjustable after fixing. Fig. 286 (*d*) gives some characteristics of these grilles. For fuller details consult makers' data.



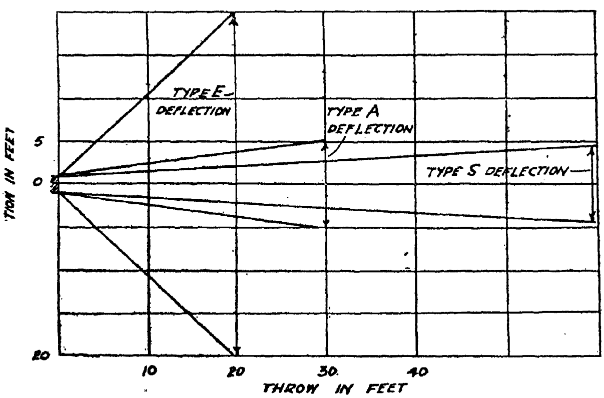
(*a*) Detail of Grille.



(*b*) Arrangement in Frame.



(*c*) Deflection Types.



(*d*) Deflection and Throw.

FIG. 286.—The '90' Grille.

Uniflow (Fig. 287)—This consists of a series of pressings to form a regular pattern, giving accurate control of direction, according to requirements. It is not adjustable after fixing.

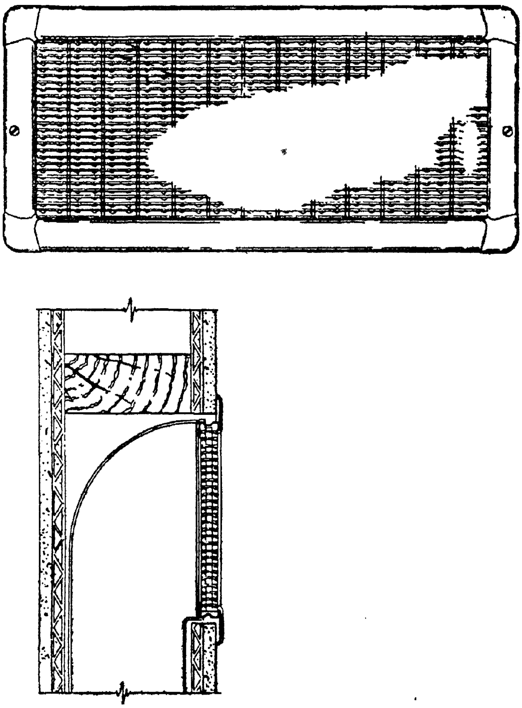


FIG. 287.—'Uniflow' Grille (Utilities Ltd.).

There are many proprietory designs of grilles following somewhat similar lines.

Ejector Inlets (Fig. 288)—This design gives high speed inlet with maximum entrainment of room air owing to the numerous thin streams. Thus

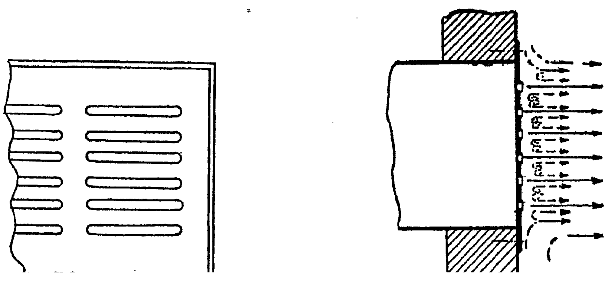


FIG. 288.—Ejector Type Grating.

for cooling, much lower temperature air may be used for the inlet, with resulting economy. With air entering 30° F. below room temperature it is stated that at a distance 3 ft. from the face of the grating it may be no more than 5° below.

Ceiling Inlets—Air cannot be introduced as in Fig. 289 directly downwards without serious risk of draughts, especially if the air is cooled.

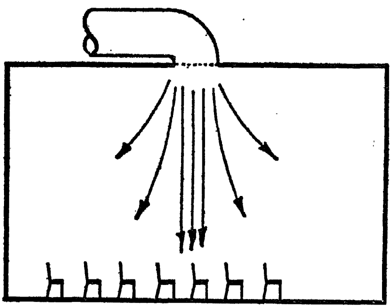


FIG. 289.—Down-draughts produced with Vertical Inlet.

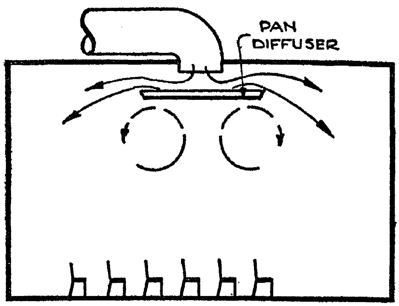


FIG. 290.—Horizontal Diffusing with Downward System.

(*a*) *Pan Diffuser*—The simplest arrangement is a pan, spaced a few inches below the inlet, and large enough to mask the opening from normal eye-level as Fig. 290. Due to the expanding periphery of radial distribution the velocity drops off rapidly and considerable air volumes can be introduced by this method without draught, whilst the inlet from the duct above may be at high velocity. Deflectors in the duct mouth may be necessary to ensure vertical flow on to the pan.

FIG. 291.—Diagram of 'Stylovent' System.

(*b*) *Stylovent*—This proprietary ceiling inlet is shown in Fig. 291. The inlet from the duct may be at 1000 ft. per min. entering vertically, but due to the deflecting rings its horizontal speed a few feet away may be as low as 300 ft./min.

(*c*) *Anemostat*—In one of its forms this appears as in Fig. 292, and has somewhat similar properties to (*b*), except that the arrangement of cones is such that room air is entrained on the underside of each plate, as shown by the arrows. Another type

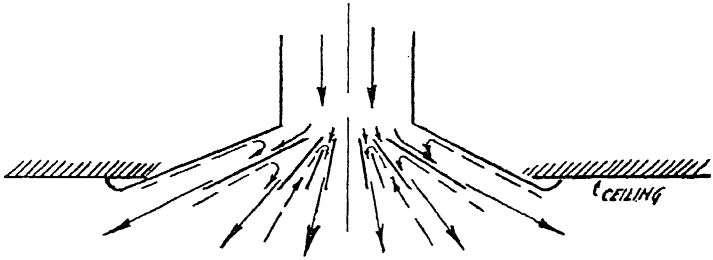


Fig. 292.—Anemostat Ceiling Diffuser. Induction Type.

is available in which the centre portion forms an extract outlet, somewhat on the same principle as the downward upward arrangement mentioned earlier (see Fig. 279).

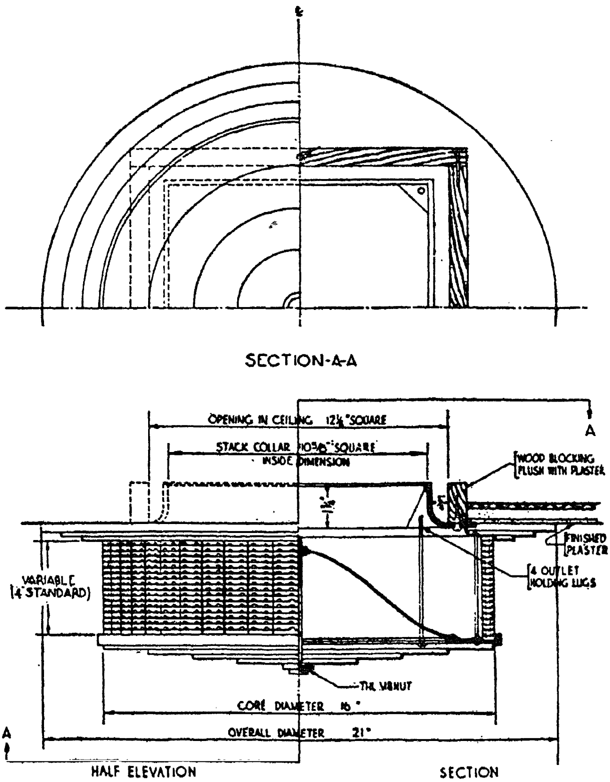


Fig. 293.—'Uniflow' Ceiling Outlet (Utilities Ltd.).

(*d*) *Uniflow Ceiling Inlet*—A ceiling diffuser using the same type of face register as the wall type (Fig. 287) is as shown in Fig. 293. It may be combined with an electric light fitting. Again the

inlet speed may be high, and the ultimate delivery speed will be satisfactorily reduced.

Nozzles—A nozzle consists of a short truncated cone, as shown in Fig. 294. Its purpose in ventilation and air-conditioning is to project the air over a considerable distance so as to give good distribution with a minimum of duct work.

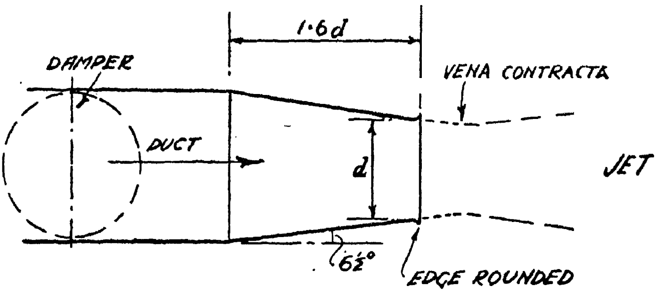


Fig. 294.—Detail of Nozzle.

The angle of slope of 6½° and ratio of length to diameter of at least 1·6 is necessary to give the highest efficiency (of static to velocity pressure) of about 92 per cent. Thus, for a velocity of 2000 ft. per min. requiring a v.p. of ·25″ w.g. (see fig. 315, p. 510), a static pressure behind the nozzle of $\cdot 25 \times \frac{100}{92} = \cdot 27''$ would be required.

Nozzles of this type may be used for any size from, say, 2″ diameter up to about 30″ diameter for very large jobs.

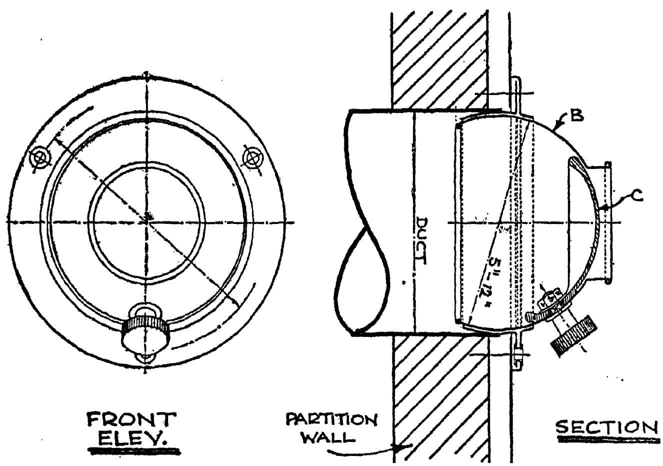


Fig. 295.—Punkah Type of High-Velocity Directional Inlet Opening. The shield *C* controls the volume delivered, and the direction of the jet is varied by moving the ball *B* in its spherical seating.

A variety of nozzle known as the 'Punkah' (Messrs. Thermotank) is shown in Fig. 295. It is adjustable for direction, and is often used in ventilation on board ship. It is made in sizes from $1\frac{1}{2}''$ to $6\frac{1}{2}''$, and is parallel-sided.

The main use of nozzles, as stated, is to cover large areas, hence it is important to know the 'length of blow' possible with any volume and velocity. Such information is not readily available. Certain formulae have been advanced, but do not appear to be borne out by tests. This is a piece of research work which will, no doubt, be tackled in earnest some day, but in the meantime the authors put forward the results of some tests carried out on nozzles at the time of the design of the Earl's Court system, described later. The results are shown graphically in Fig. 296, and are

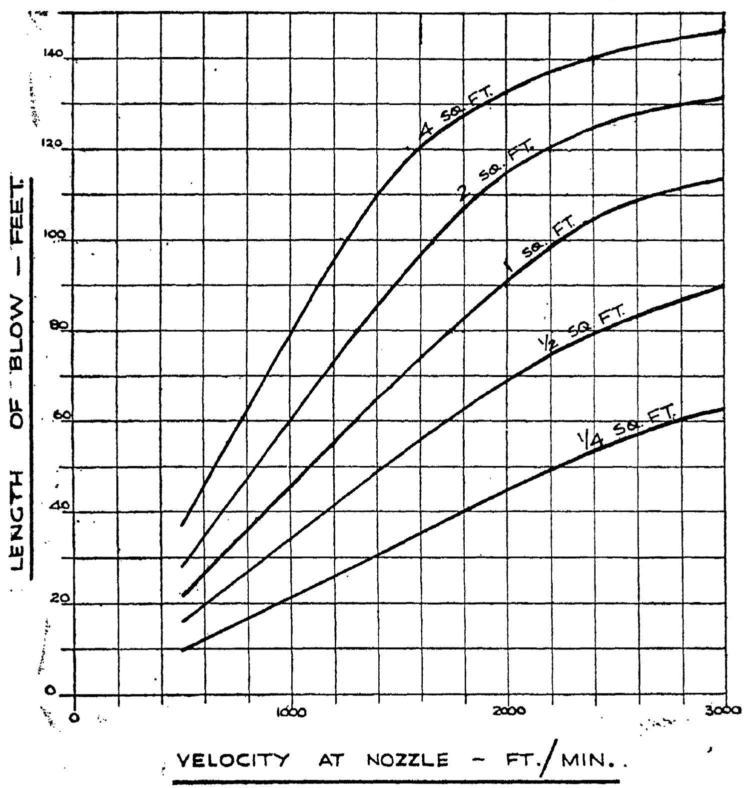
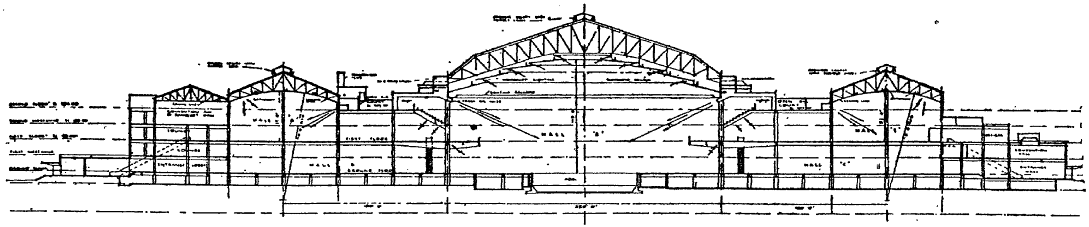
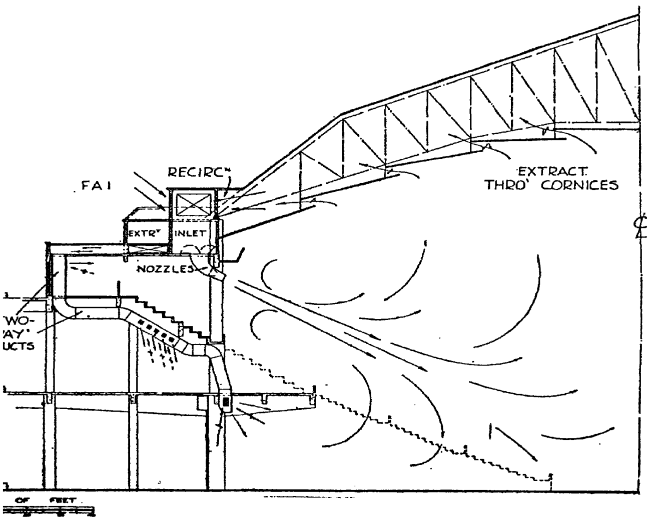


FIG. 296.—Data for Round Nozzles.

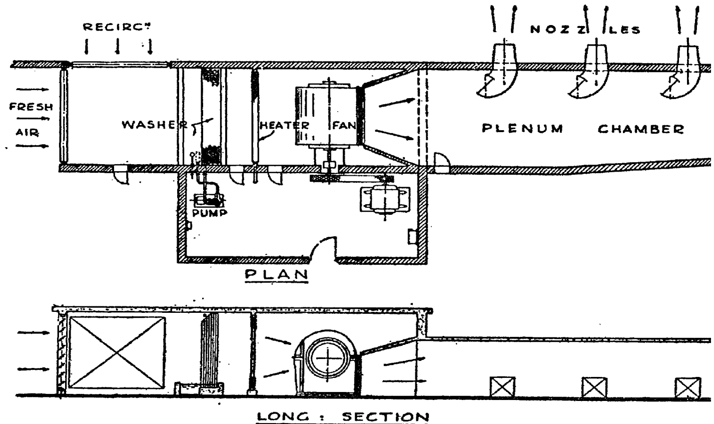
for round nozzles; other shapes would give a greater perimeter for entrainment, hence shorter throw. The length of blow is the distance from the nozzle to the plane at which the velocity has dropped to an arbitrary figure of 40 ft. per minute. It will be noted that for any given velocity the blow extends as the area of nozzle increases, though not directly *pro rata*.



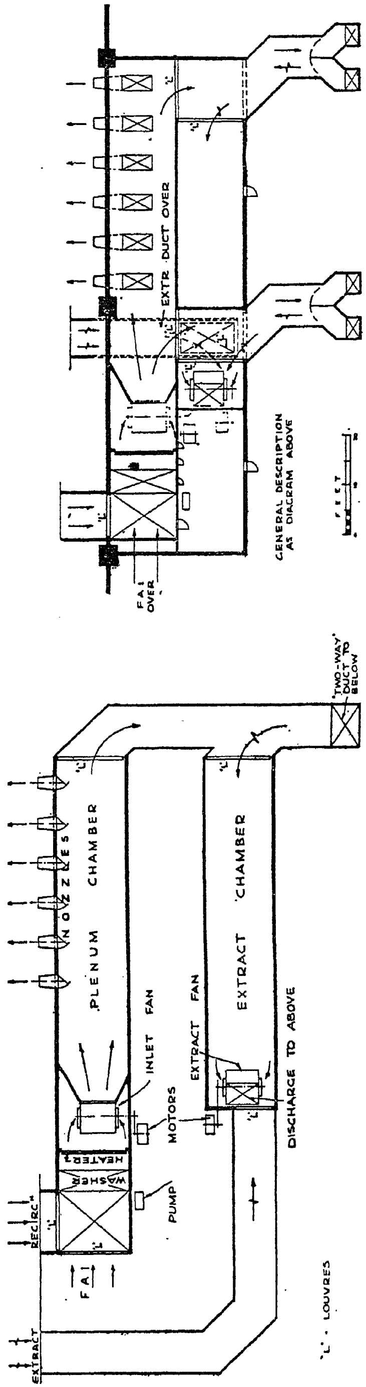
(*a*) Cross-section through Richmond Road and N.W. Entrance.



(*b*) Hall 'B' Cross-section.



(*c*) Typical Inlet Plant.



(*e*) Actual Lay-out of Plant.

(*d*) Diagram of one Hall, 'B' Plant.

Fig. 297.—Ventilation at Earl's Court Exhibition.

It was found in these tests that the sound emitted up to 2000 ft. per minute was low, but above this it increased rapidly until at 2500 ft. per minute it was too great for the use in question, though it would be admissible in certain cases.

It was also found that the fitting of a set of straightening vanes or 'egg box' in the mouth of the nozzle had the effect of reducing 'fraying' of the periphery of the jet for a considerable distance, thus increasing somewhat the effective blow.

The use of nozzles induces strong secondary currents in their vicinity, and they must therefore be kept well above head level. Excellent turbulence is created, particularly necessary for a crowded audience.

Nozzles at Earl's Court—This example is given to show the use to which nozzles can be put. Sections of the building and of the arrangement of the plant chambers are shown in Fig. 297. Plate XXVII (*a*) (facing p. 533) shows the appearance of the completed hall: the nozzles will be noted at high level on either side. Plate XXVII (*b*) and (*c*) show views of the nozzles, and of the outside of the plant rooms with intakes, exhaust ducts, etc.

The building has a cube of 50 million cu. ft., and covers 12 acres. It includes numerous halls, restaurants, kitchens, storage spaces, etc., all of which are ventilated. The main hall is 350 ft. long by 250 ft. wide by 115 ft. high; contains 11 million cu. ft.

and seats 23,000 persons. At the L.C.C. minimum of 1000 cu. ft. per occupant per hour, this called for 384,000 cu. ft. per minute. Eight inlet plants are provided, each of 48,000 c.f.m., disposed around the hall at roof level.

To provide ducting for such immense volumes of air would have proved prohibitive, hence the adoption of nozzles. These were designed to deliver at an angle as shown to give a throw of about 130 ft. Each nozzle handles 8000 cu. ft. per minute at a velocity of 2300 ft. per min., being 25 ins. diameter and requiring a static pressure of ·47 ins. w.g.

Smoke tests carried out on one of the plants showed that the air was projected nearly to the floor, and the secondary currents set up quickly diffused the smoke completely over the hall.

A complication may be mentioned by way of interest. The hall was designed to serve a dual purpose, (*a*) for seated audiences for boxing, swimming displays, spectacular scenes, etc.; (*b*) for exhibitions.

For (*a*) the downward nozzle system operates, with extract air passing out through ducts shown under the galleries in Fig. 297 (*b*), and through slots in the roof to extract fans alongside the inlet plants.

For (*b*) the L.C.C. required an upward system (to ensure removal of smoke in the event of fire). The inlet and extract plants can be changed over by dampers. The nozzles are then closed and the air is delivered through the dropping ducts leading to the lower levels under the galleries. The extract is then drawn entirely from the roof slots. The arrangement will be appreciated from a study of Fig. 297 (*d*) and (*e*). For warming up, recirculation is possible by closing the fresh air inlet dampers and opening the return dampers from the roof space.

The washer used for air filtration in this building was a modified type using scrubber plates over which water is kept running by nozzles at the top. The normal spray type washer was not possible owing to space and cost. The humidification caused by the spray type was in any event considered undesirable in this case.

The heat for the air, and the other heat requirements of the building as a whole, was supplied by the Electric Thermal Storage system shown in Plates XVII and XVIII.

This ventilation system was at the time believed to serve the largest covered area under one roof in Europe, but it certainly is the largest plant of its kind in this country.

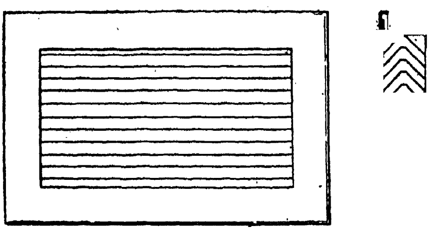


FIG. 298.—Double Louvred Natural Outlet.

Extract Gratings—The particular form of grating for extract is unimportant. It may, for instance, be of stamped steel as Fig. 283, or louvred, or any design giving the required free area. It should be remembered that dust collects on extract gratings on the outside, and close mesh or closely placed slats are undesirable as they quickly block up and impede the air flow.

Where extraction takes place naturally into a corridor, as in the office building, Fig. 239, it is necessary to use a light trap grille to avoid direct vision. Fig. 298 shows a double-louvred fitting suitable for this purpose.

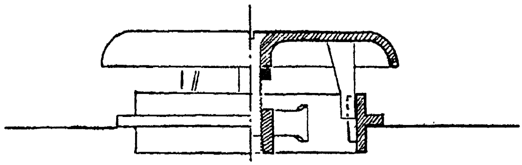


FIG. 299.—Mushroom Floor Ventilator.

Details of the mushroom ventilator and gallery-rise vent referred to earlier are shown in Figs. 299 and 300. As mentioned, these types are

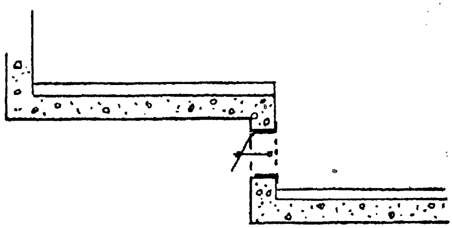


FIG. 300.—Gallery Riser Vent.

more commonly used as extracts, though they may equally be used as inlets at low velocity.

FAN TYPES AND PERFORMANCE

Static Pressure or Resistance Head—The purpose of any fan is to move air. When air is moved in a duct or through a filtering, heating, cooling or washing plant, a resistance to flow is set up.

The air is slightly compressed by the fan on its outlet side, so setting up a *static pressure* in the duct or plant. This pressure is tending to 'burst the duct', and may be read by means of a U-tube partly filled with water connected at right angles to the air stream at any point in the duct, called a 'side' gauge (Fig. 301).

On the suction side of the fan the static pressure is negative, tending to collapse the duct.

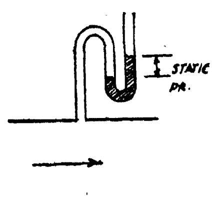


FIG. 301.—Side Gauge.

As the air proceeds along the duct from the fan its compression is gradually released until at the end of the duct open to atmosphere the air is at atmospheric pressure. This falling away of the static pressure proportionately with the length of travel is called the resistance of the duct. Similarly all obstructions, such as heaters, filters, etc., cause a loss of pressure when air is passing through them.

It should be noted that as the static pressure becomes reduced, the air in effect expands such that pressure × volume = a constant (or nearly so, as explained earlier). This expansion therefore signifies an increase in velocity of the air if the size of the duct is unchanged.

The static pressure is the same as the *Resistance Head* produced by a fan.

Velocity Pressure or Head—The fan, in addition to generating static pressure, supplies the force to accelerate the air and give it velocity. This force is termed the *velocity head*, and is proportional to the square of the velocity. It is measured by a U-tube connected to a pipe facing the direction of air flow in a duct, etc. (Fig. 302) (sometimes called a 'facing gauge'), but the pressure so measured will in addition include the static pressure, which occurs throughout the duct, as mentioned earlier. Thus, the velocity pressure alone may be found by deducting the static pressure from the total head reading, or by connecting one side of the U-tube to the facing gauge and the other to the side gauge provided the two gauges are at the same point.

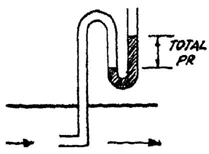


FIG. 302.—Facing Gauge.

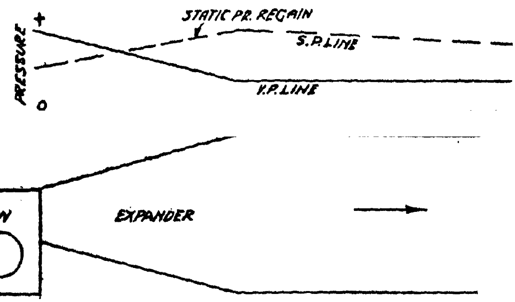


FIG. 303.

If a fan discharges into an expanding duct (Fig. 303), the velocity will obviously decrease as the distance from the fan increases, and at the same time the velocity pressure will be converted into static pressure (not at 100 per cent. efficiency, but about 75 per cent. if the expansion is sufficiently gradual). If the fan discharges into a large box (Fig. 304), from which at some point a duct connects, the fan velocity pressure will be entirely lost in eddies, and at the duct entrance must be recreated by a corresponding reduction in static pressure.

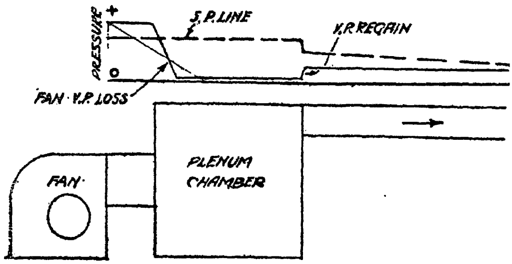


FIG. 304.

Total Head—The sum of the static and velocity heads is called the total head.

Fan Head—In all air flow considerations as affecting resistances of ducts, plant, etc., it is the static or resistance head alone which concerns us, as this is the pressure which changes with such restrictions. It is the resistance head set up by a fan which is, therefore, a criterion of its performance. The velocity head, if taken as supplementing the fan duty, may be more misleading than useful, owing to the uncertainty of friction losses which occur at points of varying velocities. The velocity head is more generally not recovered, though sufficient must remain at the duct termination to eject the air at the required speed.

Where, however, by careful design of the fan discharge expander, the velocity head is converted to resistance head (probably to the extent of about 75 per cent.), the additional head may be reckoned as augmenting the resistance head of the fan.

The head generated by a fan will be appreciated from Fig. 305.

The total fan head is defined as the algebraic difference between the mean total head at the fan outlet and the mean total head at the fan inlet.

The total head on the suction side, as will be seen from this figure, is TH_S, i.e. the negative head AO minus the velocity head equivalent to AB.

The total head TH_D on the discharge side is similarly the static head OC plus the velocity head CD.

If, as stated above, we are concerned only with the resistance head (R.H.) set up by the fan, we find that this is the difference in head of points B and C.

We can arrive at the resistance head by measuring the total head of the fan as by the U-tube 5, and deducting therefrom the velocity head as given by difference between gauges 6 and 7. Such a method is valid only if the velocity in suction and discharge ducts is the same.

The other U-tubes indicate the pressures at the various points along suction and discharge ducts, and their meaning will be apparent.

If a fan has suction ducting only, the resistance head produced for

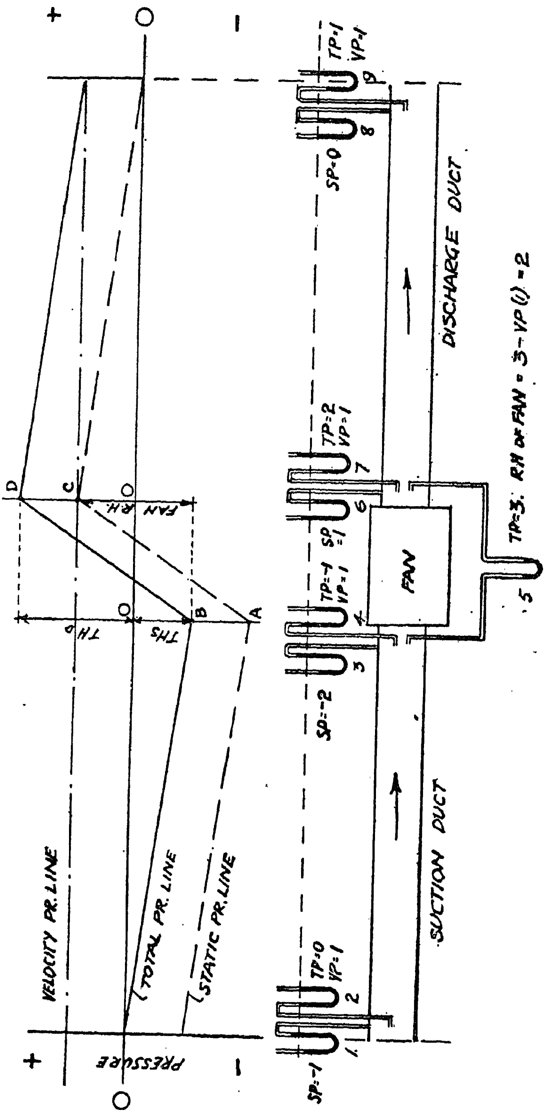


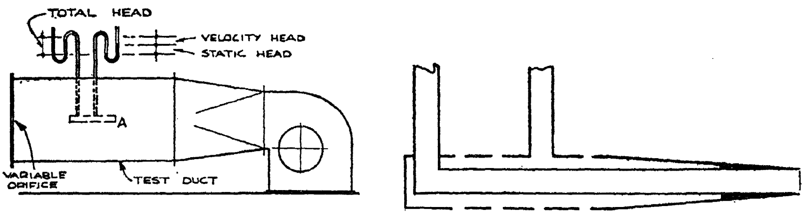
Fig. 305.

overcoming friction of ducting will be represented by OB (a negative pressure), since the discharge will be at atmospheric pressure.

Similarly, if the fan has only discharge ducting, the resistance head will be represented by OC, the suction being at atmospheric pressure.

Fan Characteristics—The comparison of the operation of fans of various types is best understood by studying their characteristic curves. For this

purpose consider a fan* connected to a duct with an adjustable orifice at the end, as in Fig. 306 (*a*). Pressures are measured by water gauges connected to a standard Pitot tube, the end of which is shown in (*b*). The perforated portion gives the static head, and the facing tube the total head.



(*a*) General Arrangement. (*b*) Detail of Pitot Tube 'A'.

FIG. 306.—Fan Test.

If the fan is running with the orifice shut no air will be delivered. Static head will be a maximum, and velocity head nil. As the orifice is opened S.H. will fall and V.H. increase until when full open S.H. will be small and V.H. a maximum. Over this range the power required to drive the fan will have increased from minimum to maximum, and perhaps will fall towards the end as the total head falls off. The horse-power may be arrived at as follows:

$$\text{Air horse-power} = \frac{\text{volume c.f.m.} \times \text{total fan head in lbs./sq. ft.}}{33{,}000}.$$

The fan water gauge reading requires to be converted to lbs. per sq. ft.

$$1'' \text{ water col.} = \frac{62{\cdot}4}{12} = 5{\cdot}2 \text{ lbs./sq. ft.}$$

$$\frac{5{\cdot}2}{33{,}000} = \frac{1}{6350};$$

$$\text{hence Air horse power} = \frac{\text{volume c.f.m.} \times \text{total fan head ins. water col.}}{6350}$$

The mechanical efficiency will be $\frac{\text{Air h.p.}}{\text{Fan h.p.}}$, and will depend on design, type of fan, speed, and proportion of full discharge.

If the water gauge used is the static head, the efficiency derived will be *static efficiency*; if the total head is used the efficiency will be *total efficiency*.

The standard air* for testing fans is taken at 60° F., 70 per cent. relative humidity, 30″ mercury barometric pressure. Any fan at constant speed will deliver a constant volume at any temperature; as the temperature varies the density will increase or decrease proportionately with the absolute temperature, hence the work done or horse-power required will vary. With increase of temperature the power will be reduced and vice versa.

* For details of standard fan testing see *Report of Fan Standardisation Committee*, I.H.V.E. Price 2s. 6d.

Similarly, decrease of atmospheric pressure (as in the case of a fan working at high altitude) will cause a reduction in horse-power and conversely.

If a fan running at a certain speed is increased to some higher speed, the resistance remaining the same, the volume will increase directly as the speed; the total head will increase in the ratio of the speeds squared; the horse-power will increase in the ratio of the speeds cubed.

Characteristic Curves—Fans are of two main types, each with sub-divisions as follows:

1. Centrifugal type:
   (*a*) Multivane, forward bladed.
   (*b*) ,, radial ,,
   (*c*) ,, backward ,,
   (*d*) Paddle wheel.
2. Propeller type:
   (*a*) Ordinary propeller or disc fan.
   (*b*) Axial flow.

The three types of runner, (*a*), (*b*) and (*c*), are shown in Fig. 307.

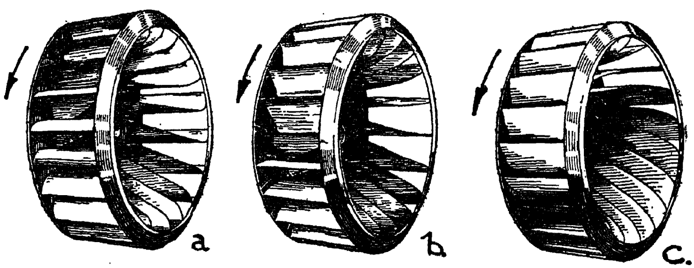
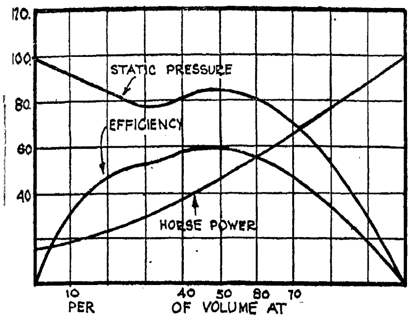


FIG. 307.—Fan Impellers: (*a*) Forward Bladed; (*b*) Radial Bladed; (*c*) Backward Bladed.

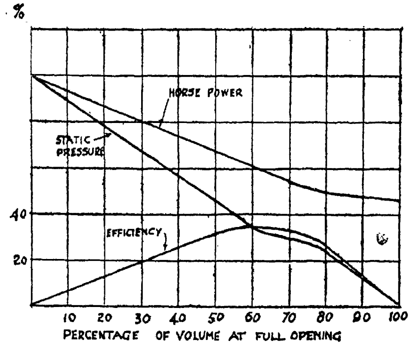
We will consider their characteristic curves and discuss their applications. The curves for resistance head or static pressure, horse-power, and efficiency (static) are drawn from tests at constant speed, as already described. The base of the curve is percentage of full opening of the orifice. The vertical scale is percentage of pressure, efficiency or horse-power.

Fig. 308 (*a*)–(*e*) gives typical curves for the five main types. Paddle-wheel fans are not given, as they are now little used in ventilating work on account of their noise, being confined chiefly to dust removal and industrial uses.

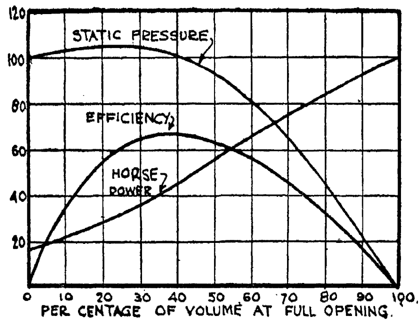
Forward Curved—It will be noted that the forward-bladed centrifugal fan most commonly used in ventilation reaches a maximum efficiency at about 50 per cent. opening, where at the same time the static pressure is fairly high. Fans are generally selected to work near this point. It will also be observed that the horse-power curve rises continuously. Thus, if in a duct system the resistance is less than calculated, the air delivered will be more and the horse-power more, which will lead to overloading of the motor.



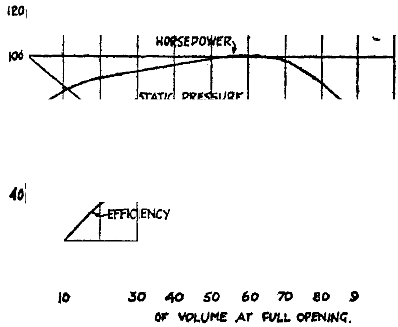
(*a*) Forward Bladed.



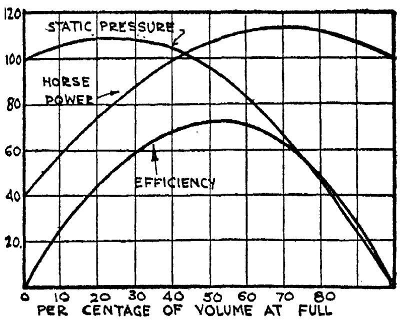
Propeller Fan.



(*b*) Radial Bladed.



(*e*) Axial Flow Fan (Airscrew).



(*c*) Backward Bladed.

FIG. 308.—Characteristic Curves of Centrifugal, Propeller and Axial Flow Fans. The vertical scale represents:

Percentage efficiency
,, horse power (over range).
,, static pressure (outlet closed = 100%).

Backward Curved—This type of fan runs at a higher speed to achieve the same output as a forward curved. The efficiency reaches a maximum at about the same point, and the horse-power, after reaching a peak, begins to fall. This is called a *self-limiting characteristic*, and means that if the motor is installed large enough to cover this peak it cannot be overloaded. This is often useful in cases where the pressure is variable or indeterminate. The

pressure curve is smooth without the dip of the backward curved; for this reason this kind of fan is to be preferred where two fans are working in parallel. The forward curved type under such conditions is apt to hunt from one peak to the next, so that one fan may take more than its share of the load and the other much less.

Radial—This type is in effect similar to a backward curved in some respects, but has not the power-limiting characteristic. It is not commonly used, but some makers standardize on it.

Ordinary Propeller—From the curves it will be observed how the pressure falls away continuously, and the static efficiency reaches but a low figure. Thus, this type is unsuitable where any considerable run of ducting is used. To generate any appreciable water gauge its speed becomes unduly high, and hence the fan is noisy. Its main purpose is for free air discharge where its velocity curve would rise towards a maximum at full opening. It should be noted that the horse-power is a maximum at closed discharge, and as the motors supplied with these fans are not usually rated to work at such a condition the closing or baffling of the discharge or suction may cause overloading.

Axial Flow—This type shows a great improvement over the ordinary propeller fan, both as regards efficiency and pressure. The horse-power curve is self-limiting. Hence, these fans may safely be used in conjunction with a system of ductwork, being often more convenient than a centrifugal, particularly for exhausting.

Fan Arrangements and Drives—Centrifugal fans may be *open* or *cased*. When open they can only be used for exhausting, and the discharge is tangential from the perimeter of the impeller as might be suitable in a large roof turret.

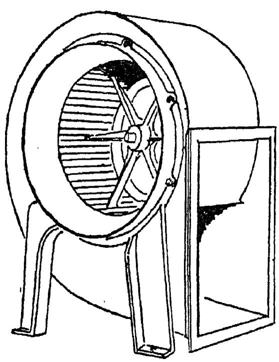


FIG. 309.—Single Inlet Centrifugal Fan.

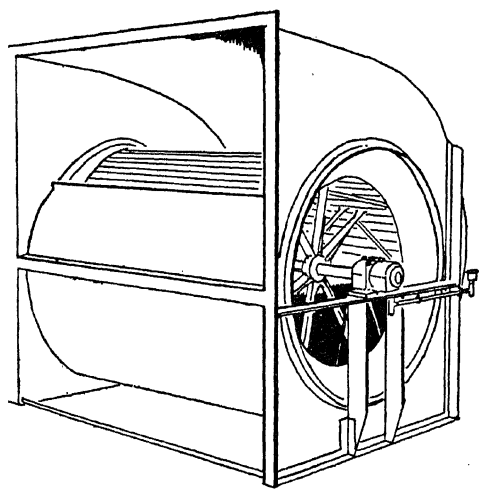


FIG. 310.—Double Inlet Centrifugal Fan.

The usual arrangement is the cased type, and the suction is then either on one side, as in Fig. 309 (single inlet), or both sides as Fig. 310 (double inlet). The single inlet is the more usual. The double inlet double-width

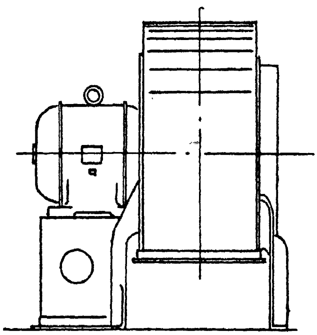


FIG. 311.—Centrifugal Fan with Close-coupled Motor.

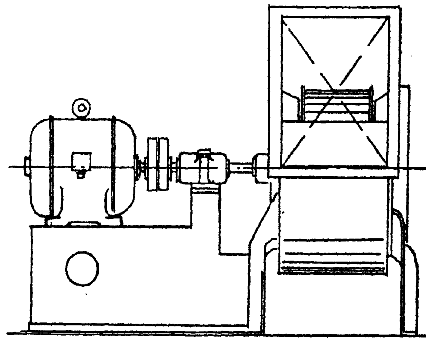


FIG. 312.—Centrifugal Fan with Flexible Coupling to Motor.

fan is useful where large volumes are concerned, as it gives double the capacity of the single inlet with the same height of casing.

It will be noted, for instance, from the lay-out of the Earl's Court plant given in Fig. 297, that double inlet fans were used in order to economize space.

Fans are now invariably driven by electric motor. Fig. 311 shows a typical arrangement with the fan impeller mounted on a shaft extension of the motor. This is a compact arrangement, but generally used for small or medium-sized fans only.

Fig. 312 shows a motor direct-coupled to a fan with a flexible coupling. The fan shaft runs in its own bearings independently of the motor. This is obviously to be preferred for heavy duty and for large fans. The motor can be removed and replaced without affecting the fan.

Fig. 313 shows a motor driving a fan by Vee-rubber belts. This arrangement has the great advantage that the motor speed may be that best suited

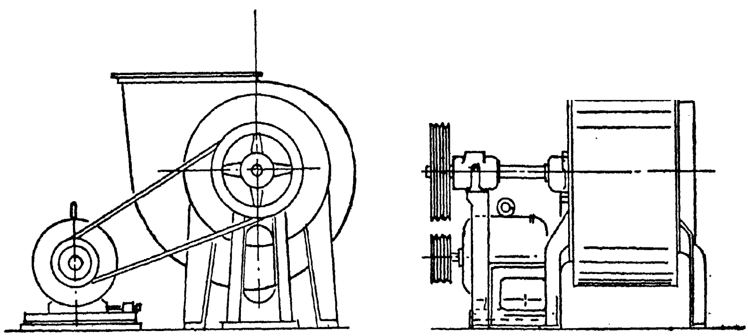


FIG. 313.—Centrifugal Fan with Vee Rope Belt Drive (Matthews & Yates).

to the current, such as 960 r.p.m. or 1440 r.p.m., whilst the fan speed is that best suited to the fan. A further advantage is that if on testing it is found that the resistance of the system is less or more than allowed for, the fan duty can be corrected by merely changing the pulleys.

It will be noted that in the illustrations different positions of the discharge openings in relation to the suction eye of the fan in each case are given. It is possible to obtain a fan with its discharge at any angle of discharge, vertical, horizontal top, horizontal bottom, downwards, and intermediately at an angle of 45°.

Fan Noise—Centrifugal fans may be used up to relatively high pressures, such as 6″ or 8″ w.g., but for normal ventilating use the range is confined to between $\frac{1}{8}$″ and 3″ w.g. Above $1\frac{1}{2}$″ w.g. noise becomes a difficulty, and is only overcome by isolation and insulation.

The following notes cover the methods generally adopted to reduce fan noise:

(*a*) Fans bearings and motors should be isolated from the building by cork or other anti-vibration material. This is often inserted between the concrete base and the floor of the plant room.
(*b*) The fan suction and discharge should be connected to the ducting by means of sailcloth connections.
(*c*) Motors of super-silent type should be used.
(*d*) Sleeve type ring-oiled bearings are preferable to ball bearings, both for fan and motor.

For quiet running a large fan at a slow speed is generally better than a small fan at a high speed. A useful criterion is the outlet velocity. Recommended outlet velocities for silent conditions, as when ventilating an auditorium, living space, etc., lie between 1300 and 1400 ft. per min.; where slight noise is not objectionable, up to 1600 ft. per min. may be used, and in more noisy locations up to 2000 ft. per min. Speeds above this are generally only used in industrial applications.

Axial Flow Fan Arrangements—This type of fan, it is stated, can be designed to give static pressures up to 2″ w.g. within the limits of quiet running. The maximum peripheral speed should not normally exceed 7500 ft. per min. for this condition.

These fans can be built in one, two or three stages, to obtain increased pressure, the volume remaining the same. Alternatively they may have two blades made counter-rotating.

The Airscrew Fans, illustrated in Fig. 314 (*a*)–(*e*), have blades of specially treated wood, similar to an aeroplane propeller; these are shaped to aerofoil section (see (*a*)), and may be designed for any angle and any number of blades from two to twelve. Wood is light, and easily shaped to smooth edges, thus avoiding the sharp cutting action and noise associated with sheet metal blades.

(*b*) shows a detail of the arrangement with eight blades, driven from direct-coupled motor in the duct. (*c*) shows drive by external motor with Vee belt, suitable where steam or fumes have to be removed. (*d*) shows a

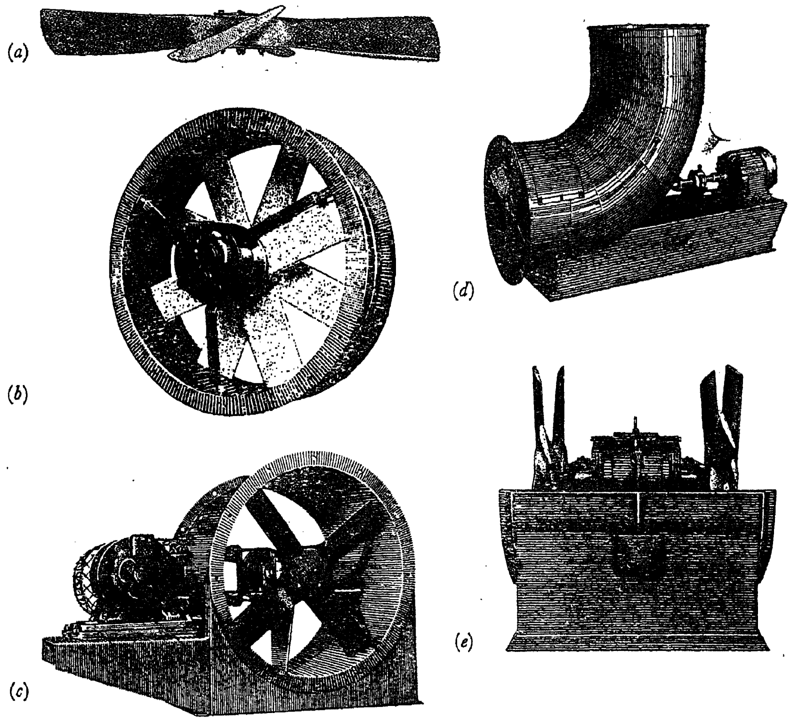


FIG. 314.—Axial Flow Fans (Airscrew Ltd., Weybridge).

fan in a bend driven by direct-coupled external motor; this type is described as an *acid-bend* fan, bend and fan being treated with anticorrosive protection. This is suitable for many ventilation applications in chemical plants, etc., but it may also be used for extraction, say, of kitchen exhaust fumes, laundry steam, etc. (*e*) shows a two-stage fan driven by a common motor, mounted in a duct, the top half of which has been removed.

Other makes of axial flow fans use cast aluminium blades, and include streamlined nose and tail pieces to the motors. Such a type is the 'Aeroto'.

Enough has no doubt been said to indicate the wide advance which this type of fan has made over the old plain propeller type.

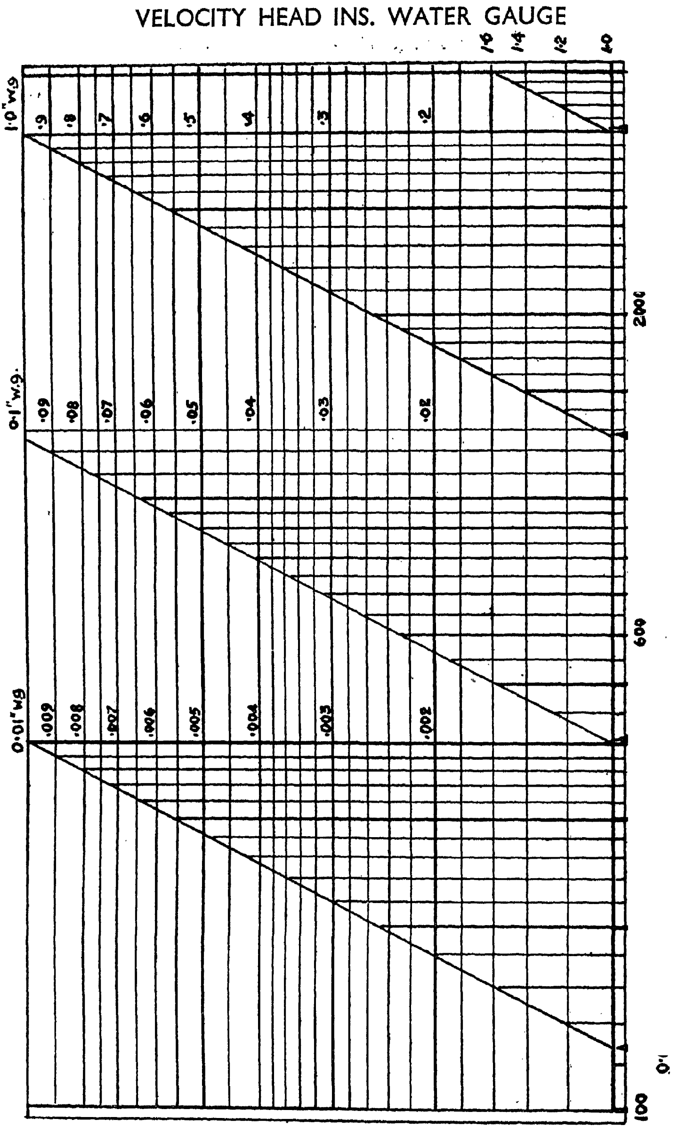
Fan Duties—The range of fan types, speeds, pressures, and volumes is too great for any indication to be given here of sizes, duties, horse-powers, etc., or of the problem of motor types suitable for fan drives.

Enquiries to fan-makers should give the fullest information possible

about any system, as there are many hidden points to be watched in the selection of fans which render mere catalogue reference insufficient.

MEASUREMENT OF AIR FLOW

Pitot Tube—Reference has been made to the Pitot tube as a means of measuring static and velocity pressure. It may be used in its latter capacity to determine the velocity in a duct.



315.—Veloc related to Air S Standard Air.

The relationship between speed and pressure is:

$$V = 1096{\cdot}5\sqrt{\frac{h}{\rho}},$$

where V = air vel. in ft./min.

h = velocity head in inches w.g. (i.e. the potential-energy equivalent of the velocity).

ρ = density of air in lbs./cu. ft.

For air at 60° F. 60 per cent. R.H. and 1000 m.b. pressure,*

$$\rho = {\cdot}0745 \text{ lbs./cu. ft.}$$

$\therefore$ for this condition, $V = 4019\sqrt{h}$.

. 315 gives this relationship graphically for speeds encountered in ventilation work.

Micromanometer—The pressures at low velocities are slight, as will be noted, and for the purpose of reading them an ordinary U-tube or inclined gauge is too insensitive. The micromanometer, of which one type is shown in Fig. 316, is therefore necessary. In this an extended U-tube is used,

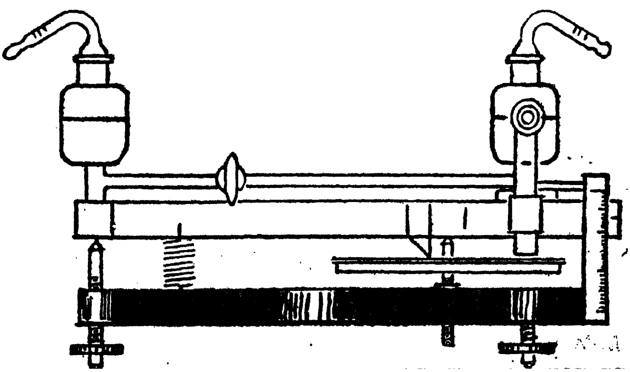


FIG. 316.—Tilting Micromanometer.

being tilted by a micro-adjustment. The level is viewed through the magnifying eyepiece against a crosswire. The coarse reading is taken on the side scale, and the fine reading on the rotating dial, one revolution of which corresponds to one division of the scale. Other still more sensitive instruments are made.

When measuring air speeds in a duct, however, the velocity varies across the duct, being greatest at the centre and least at the periphery. Thus, in a round duct it is necessary to take readings at a number of

* This is the standard air used as a basis for the *I.H.V.E. Hygrometric Tables and Duct Sizing.* It is different from the Standard Air for Fan Testing, see p. 503.

points in concentric rings of approximately equal area. The average speed multiplied by the area of the duct will give the volume of air passing. In the case of a rectangular duct the method is similar, except that the duct is divided into equal rectangles.

Anemometer—The anemometer (Fig. 317) measures air speed by vanes which revolve as the air impinges on them. The dials, which are calibrated in feet, serve only to count the revolutions over a given time, such as a minute, taken by a stop-watch.

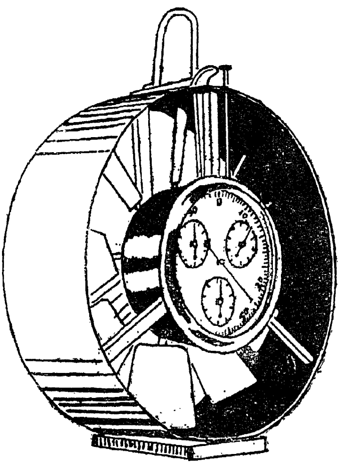


FIG. 317.—Vane Anemometer.

The lever at the top enables the gearing to be engaged and disengaged from the vane at the start and finish of the time. The knob shown on top is depressed to cancel the reading and return the dials to zero.

The instrument requires to be calibrated periodically, and a correction factor applied. The standard instrument is too insensitive for use below about 100 ft. per min., but other slow-speed types may be used down to about 30 ft. per min. Above about 2500 ft. per min. a high speed type is used.

For general work it is a useful instrument, chiefly for measuring the air speed from or to gratings, etc. In such case the instrument is placed about one inch from the grille and the speed is multiplied by the whole face area (regardless of grating-free area) to obtain the volume in cu. ft. per min. Readings are taken at various points on the grille and averaged. It is not so useful for measurements in ducts, as the anemometer has to be introduced through a hole in the duct wall, and it is then difficult to manipulate.

It must be admitted that the anemometer requires very careful handling, and in the hands of an unskilled operator entirely erroneous results can be obtained, particularly when measuring over a large grille.

The Velometer—A more recently-developed instrument, the 'Velometer', is shown in Fig. 318. It relies on the speed of air to deflect a shutter against a spring. The shutter causes the needle to move over the sector-shaped dial, reading direct in feet per minute. It requires no timing.

When used direct, as in the illustration, for measuring speeds from or to gratings at low velocities, it reads from zero to 300 ft. per min. Higher ranges of speeds, 0–3000 and 1000–6000 ft. per min., are covered by means of an adaptor screwed into the aperture at one end. This may have a rubber tube connection for reading at a distance, which, in turn, may be used with various special mouthpieces fitted to the end of the rubber tube

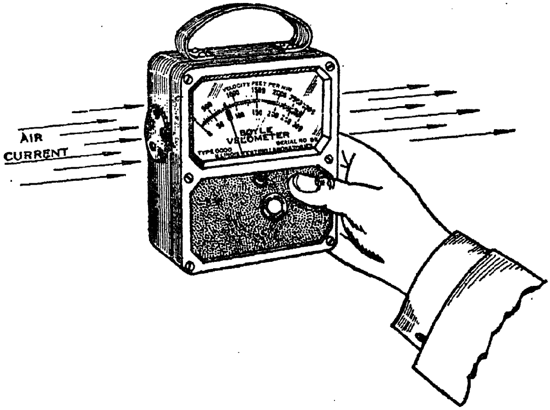


FIG. 318.—Velometer (Metropolitan Vickers).

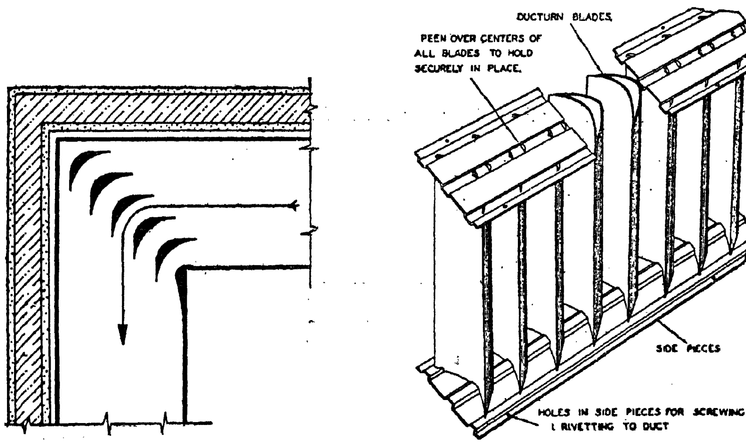
to read velocities in ducts, etc. The same instrument may be used in place of a pitot tube to give static pressure readings.

Altogether this is a most useful instrument for giving a quick check on adjustments or for exploring velocities over an area.

AIR DUCTS

Ducts for ventilation and air-conditioning are commonly constructed either of galvanized sheet steel or of 'Builders' Work'. The principles to be followed in the design of such ducts are:

(*a*) Avoidance of sudden restrictions or enlargements, or any arrangement producing abrupt changes of velocity.



. 319.—Standard Duct Turn.

(*b*) Bends to be kept to a minimum, but where required should have a centre-line radius not less than one and a half times the diameter or width of duct at the bend. Alternatively, deflectors should be fitted into the duct, of the type shown in Fig. 319, which permit of right-angled turns being used with much reduced pressure loss, and reduction of noise.

(*c*) Where branches occur, they should be taken off at a gradual angle before turning.

(*d*) Sharp edges should be avoided, as these will be the cause of noise which may travel a considerable distance through the duct system.

(*e*) Rectangular ducts should be as nearly as possible square, and in any case the ratio of width to height should not exceed 5 : 1 (preferably 3 : 1).

Galvanized Steel Ducts are formed in short lengths out of flat sheets by riveting or bolting, and the lengths are erected in position with slip-joints, or angle-ring joints bolted together. The detailed construction of ducts is a highly specialized technique based on experience.

Circular ducts are preferable in that they contain less metal for a given area than any other shape. They can also be made thinner, and for both these reasons are cheaper than rectangular ducts. Circular ducts are, however, generally impracticable except in factories, roof-spaces, basements, etc., where adequate room is available.

Rectangular ducts are liable to drumming, especially where of fair size. For this reason they should be stiffened, or an alternative method is to punch the plates with a 'diamond break' as in Fig. 320.

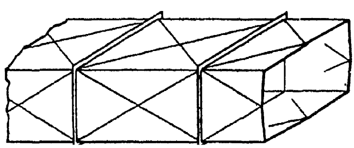


FIG. 320.—Diamond-break Stiffening of Duct Walls.

Builders' Work Ducts—It is often possible, in the design of a building, to construct the ducts as part of the fabric. Where this can be done it has the following advantages:

(*a*) Rigidity, and hence reduction of noise.

(*b*) Permanence.

(*c*) Cheapness, in that the space very often exists in any case, and only requires to be made smooth and air-tight.

(*d*) Reduction of heat gains and losses, due to the heavy construction as compared with metal.

(*e*) Accessibility for cleaning, etc.

A disadvantage is the 'flywheel' effect of heavy constructions and hence for applications where careful control of the air temperature is required, such ducts must be insulated *on the inside.*

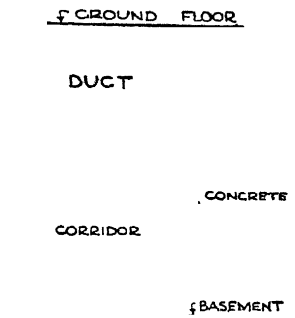


FIG. 321.—Main Duct over Basement Corridor.

Fig. 321 shows an example of a duct constructed over a basement corridor and serving as a main distribution trunk from the plant to rising shafts about the building. Fig. 322 shows a case in which the ducts were constructed under the basement floor to serve rising shafts in the same way.

Ducts of Other Materials—Other materials used for ducts are:

(*a*) *Asbestos Cement,* which is performed in sections and erected like a metal duct. It is more commonly used for fume extraction owing to its immunity from corrosion.

(*b*) '*Metaline*',* which is a fibrous-plaster material reinforced with scrim. It can be moulded to any desired shape, and can be cut and adapted at site, and jointed to form a smooth continuous duct. It is especially suitable for insulated ducts, as cork insulation can be moulded in with the material.

(*c*) *Copper, Aluminium, etc.*—Ducts constructed of these materials are used in special cases where extreme permanence is desired, or where special corrosion problems arise.

(*d*) *Welded Sheet Steel* (ungalvanized) is mainly used in industrial work, or where the air-tightness of the joints is of great importance.

Noise in Duct Systems—Noise may come from the plant (i.e. fan, washer, pump, etc.) and be conveyed through the ducts, or may be caused by the ducts themselves. It is unfortunately not always possible to eradicate noise completely at its source, and where a high degree of silence is required, as in the case of concert halls, film studios, private offices, etc., it is generally necessary to insulate the ducts, or portions of them, so that such noises as do arise are not conveyed to the rooms served.

* Messrs. De Jong & Sons Ltd.

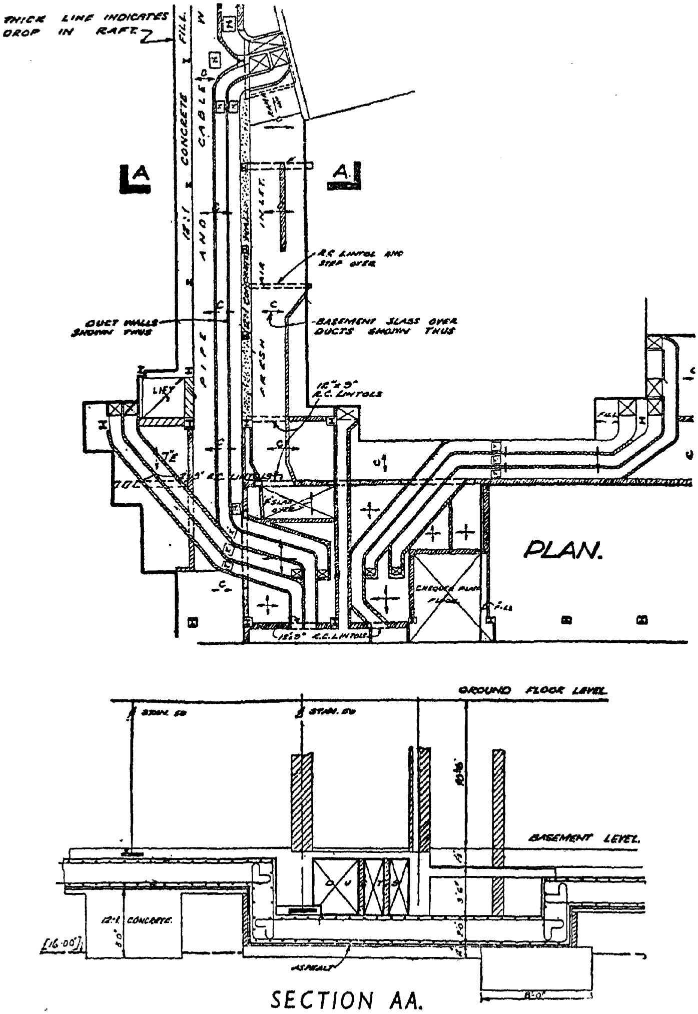


FIG. 322.—Example of Ducts formed in Space under Basement Floor.

One method is to line the ducts with a sound absorbent material such as 'Pax-felt',* eel-grass blankets, or other similar material (which should be fire-resisting), and which may be fixed as shown in Fig. 323. It is in some cases sufficient to line only a length of about ten diameters, near the fan, and in other cases to line only the bends, etc., where air impinges.

In addition to noise originating in the system itself, there is also a danger of noise from one room being carried to the next along the ducts.

* Messrs. Newalls Insulation Co.

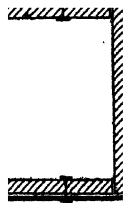
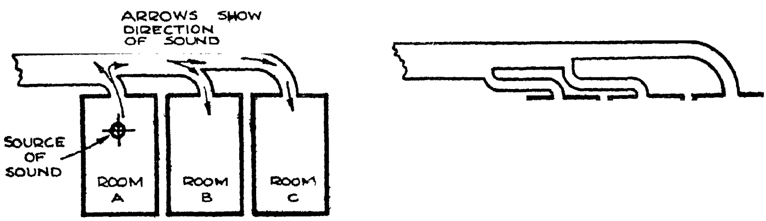


FIG. 323.—Method of Fixing Insulation in Metal Duct.

This will be apparent from Fig. 324 (*a*). It may be overcome by running separate ducts to the rooms as in (*b*), each branch duct being preferably insulated.



(*a*) FIG. 324. (*b*)

Where silence has to be achieved with ducts of a short length, it is necessary to use a sound-absorber of honey-comb formation, of the type shown in Fig. 325, formed in an expansion of the duct so as to allow the same free area as elsewhere. The minimum length of the absorber is about 8 feet.

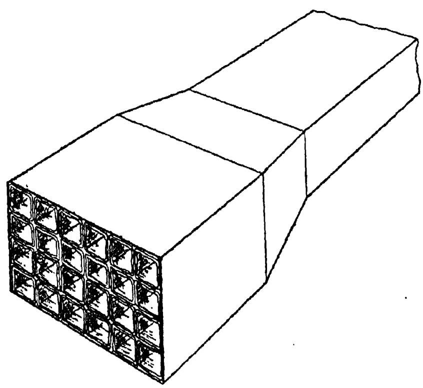


FIG. 325.—Sound-absorber for Use in Ducts, using Eel-grass Honeycombs.

Insulation of Ducts—Apart from the question of sound, insulation of ducts carrying cooled air is essential, as otherwise much of the cooling effect

is lost. For this purpose ducts may be insulated as for sound, on the inside, in which case the insulation serves a dual purpose, or in the case of metal ducts, on the outside, which is generally more convenient. Such insulation would normally take the form of 1″ cork where inside the building, and 2″ cork where outside, or the equivalent in glass fibre or other material.

Cooling costs much more than heating, hence insulation is more remunerative. To give, say, 1,000,000 B.T.U.'s of cooling a refrigerating machine motor of about 150 h.p. would be called for, with a running cost at 1d. a unit of about 12s. 6d. per hour. The same amount of heating produced by coal at 30s. a ton in a boiler would cost 2s. 1d., or one-sixth of the cost of cooling.

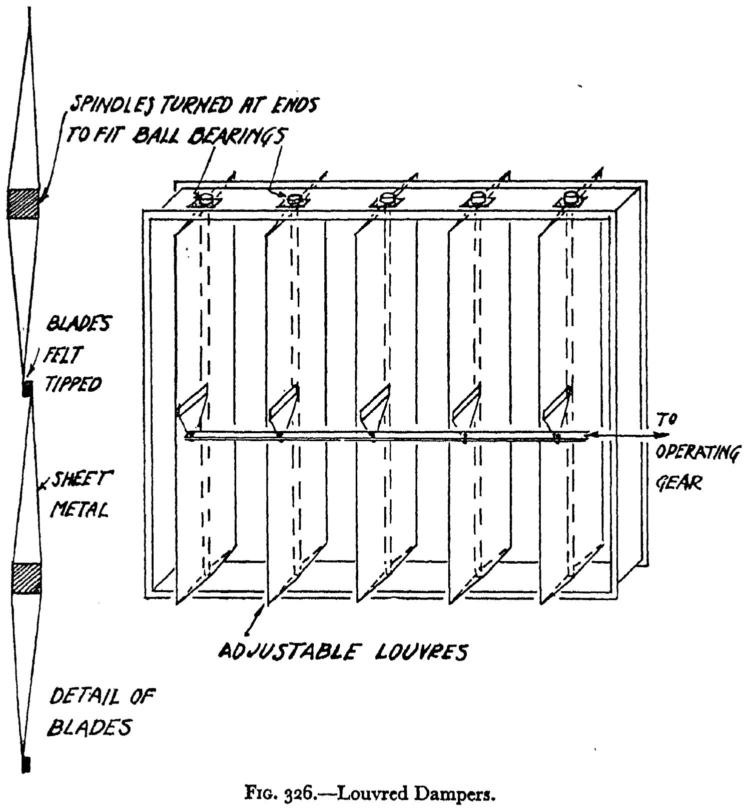


Fig. 326.—Louvred Dampers.

Dampers in Ducts—Control of the air volume is effected by means of dampers, which are either:

(*a*) Permanently set when the installation is tested, to give the designed volumes in each branch. These are commonly of butterfly or sliding type with some form of locking device.

(*b*) 'Controllable' dampers for use in air-conditioned installations, and adjusted, either manually or automatically, to suit varying conditions. Such dampers are generally of louvred type as in Fig. 326.

Where tight shut-off is essential, it is necessary for the dampers to be felt-tipped and to close on to a felted frame.

Duct Sizing—The most convenient method of sizing ducts is on the *Equal Pressure-loss* basis. For this purpose use may be made of Figs. 327–9 and Tables LXXXII and LXXXIII.

The following procedure is then adopted:

(1) On the plan of the duct system it is necessary to mark the volume of air to be delivered or exhausted at each outlet. These must be totalled back to the fan, including the sums brought in at each branch. These volumes are conveniently kept in cubic feet per minute (c.f.m.).

(2) Establish the maximum velocity in the main duct leaving or entering the fan according to the type of building, etc., from Table LXXXII. The velocities given in this table are arbitrary. The higher the velocity the greater the noise, hence low velocities are desirable for buildings where silence is required. Some experience is necessary in interpreting this aspect of the matter.

(3) From Fig. 327 select the velocity line for the main duct, and find the point of intersection with the air volume carried in the main. The vertical line passing through this point will give the resistance in inches of water gauge per 100 ft. run of duct by reference to the bottom scale, and at the same time the diameter of main duct, assumed to be circular.

(4) The sizes of the subsequent sections of the main duct on to the end are then read off at the point of intersection of the appropriate horizontal volume lines with the same vertical resistance line as for (3). It will be observed that as the volume reduces, the velocity also becomes less.

(5) The total resistance of the main duct is then calculated by tabulating thus:

(*a*) Section (Ref. Letter or No.)	(*b*) Volume c.f.m.	(*c*) Duct Diameter ins.	(*d*) Length ft.	(*e*) Resistance per 100 ft. from Chart " W.G.	(*f*) Resistance for Length (*d*) " W.G.	(*g*) Single Resistances " W.G.	(*h*) Total for Section " W.G.	(*i*) Progressive Total " W.G.

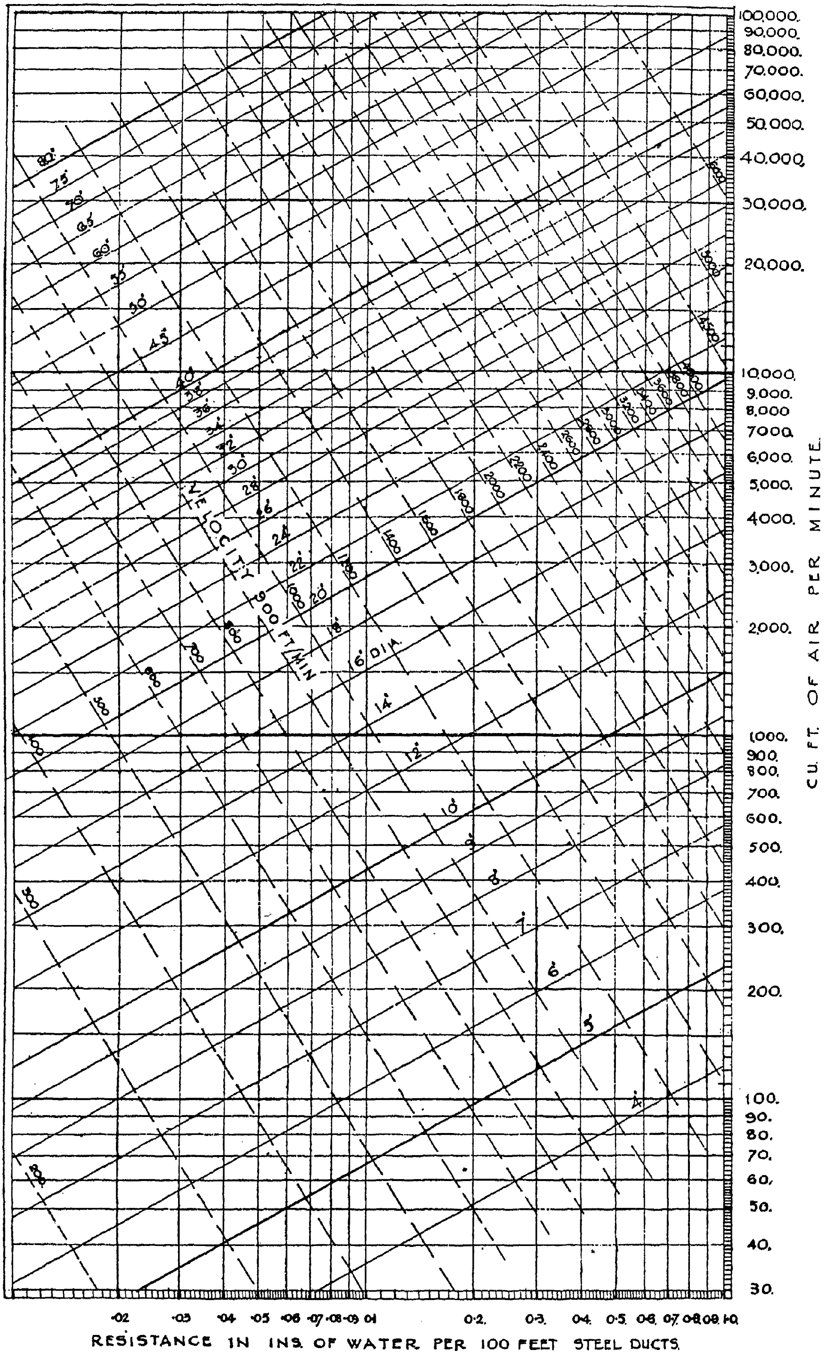


FIG. 327.—Duct Sizing Chart (Based on *I.H.V.E.* data).

The values of single resistances (for bends, etc.), column (g), are obtained on the velocity pressure (V.P.) method. The pressure corresponding to any velocity is taken from Fig. 315. The loss in terms of V.P. for each particular bend, reducer, etc., is taken from Fig. 329.

(6) The branches are then dealt with thus:

Pressure available for branch = (total resistance of main duct) − (resistance from fan to branch, column (i)).

The length of the branch to its end is then measured and the

$$\frac{\text{Pressure} \times 100}{\text{length}} = \text{resistance per 100 ft. for the branch.}$$

With this new resistance, duct sizes for each volume are read off (Fig. 327) as before.

If this gives too high a velocity, a lower resistance must be selected, and the surplus pressure can then only be absorbed by dampering.

(7) Sizes for the whole system in terms of circular ducts will then be marked on the plan.

If rectangular ducts are used instead of round, the equivalent sizes to give equal resistance per 100 ft. may be taken from Fig. 328. The rectangular sizes may be selected to suit the positions which the ducts must occupy.

The figure is based on the formula

$$D = 1{\cdot}265\left(\frac{A^3B^3}{A+B}\right)^{\frac{1}{5}},$$

where A and B are the dimension of a rectangular duct.

D is the diameter of the circular duct which has the same resistance per unit length when carrying the same air volume as the rectangular duct.

It will be noted that the values of A, B and D are given from 1 to 10 only. These values may be in any units, i.e. one inch, ten inches, feet, etc., provided the unit is used consistently for all three variables. Thus a duct $72'' \times 84''$ may be converted to a circular duct (of $85''$ diameter) by interpolating between the lines, or equally by reading on the intersection of the horizontal 6·0 and the sloping 7·0 lines in feet ($D = 7{\cdot}1'$).

(8) Where ducts are made of other materials, the resistance will be less or more than that of galvanized steel ducts as given in Table LXXXIII.

(9) Ducts of galvanized sheet steel are priced in tons weight. Table LXXXIV gives standard gauges of metal used for ducts and weight per sq. foot of surface.

Fan Pressure—The static water gauge or resistance head to be set up by the fan is the sum of all resistances to air flow throughout the system as follows:

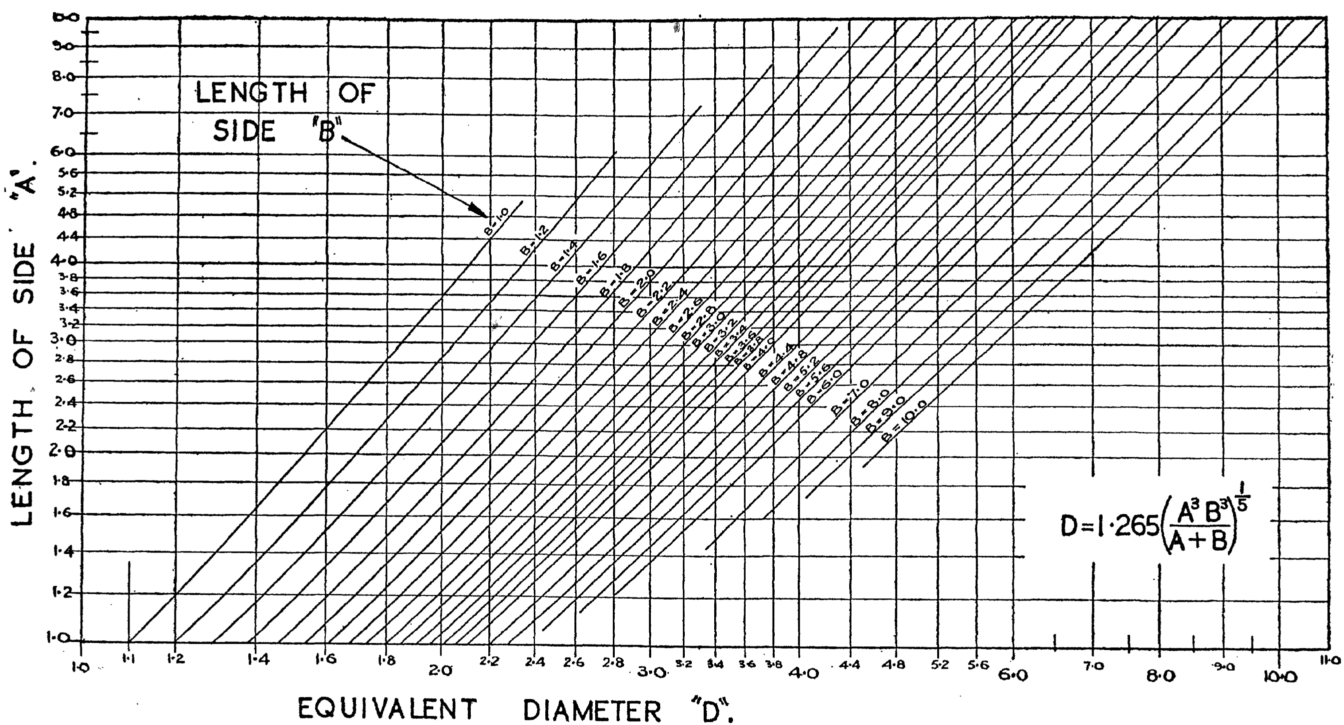


FIG. 328.—Equivalent Circular and Rectangular Duct Sizes for Equal Friction and Volume.

	Resistance, Ins. W.G.
(1) Fresh air intake louvres or grille (estimate on V.P. method, Fig. 329) - -	
(2) Fresh air intake duct (size and resistance established as for other ducts). This must include loss in single resistances as before - - - - -	
(3) Resistance of:	
(a) air filter if any - - - - - - - - - -	
(b) heater or heaters - - - - - - - - - -	
(c) washer if any - - - - - - - - - -	
(d) cooling coils if direct expansion cooling - - - - - - (The above will be determined from makers' data.)	
(4) Convergence of fan suction - - - - - - - - - (Use V.P. method.)	
(5) Divergence of fan delivery - - - - - - - - - (Use V.P. method.)	
(6) Changes of section, if any, through plant - - - - - - - (Use V.P. method.)	
(7) Resistance of duct system estimated as already tabulated - - - -	
(8) Resistance of final outlet grille - - - - - - - -	
Resistance Head of Fan - - - - - - -	

Extract System—Exactly the same summation of resistance is arrived at for an extract system as for an inlet. Items (3) and (6) in the above calculation will disappear, and in place of item (1) must be included the resistance of the discharge duct and cowl or other outlet.

TABLE LXXXII

MAXIMUM DUCT VELOCITIES RECOMMENDED FOR VARIOUS TYPES OF BUILDINGS

	Main Ducts and Shafts	Branch Ducts
Hospitals, Concert Halls, Theatres, Libraries, Film Studios, etc.. where silence essential - - - - - - - - -	1000	800
Cinemas, Restaurants, Assembly Halls, etc. - - - - - -	1500	1000
General Offices, Dance Halls, Shops, Exhibition Halls, etc. - -	1800	1200
Factories, Workshops, etc., where noise unimportant - - -	2000 to 2500	1500

TABLE LXXXIII

FACTORS FOR DUCTS OF OTHER MATERIALS

(Basis: Taking resistance from Fig. 327 as 1·0 for galvanized steel, the resistance should be multiplied by the appropriate factor given below.)

Smooth Copper, Aluminium, or Enamelled Plaster - - -	0·8
Smooth Cement - - - - - - - - - -	1·2
Rough Concrete or Good Brick - - - - - - -	1·5
Rough Brick or Pre-cast Concrete - - - - - -	2·0

Fig. 329.

Velocity Pressure Loss Factors for Single Duct Resistances.

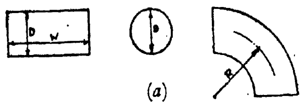
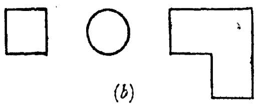
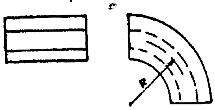
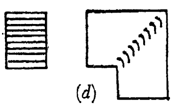
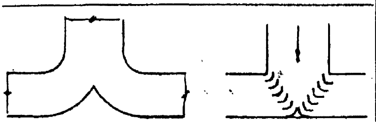
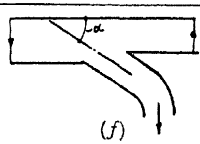
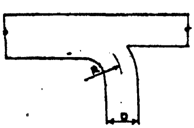
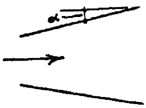
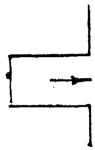
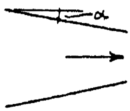
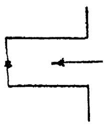
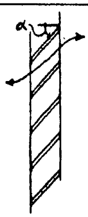
	Duct Element	Circular			Rectangular	
		$\frac{R}{D}$	90°	135°	$\frac{W}{D}$ = ·5	$\frac{W}{D}$ = 1 to 4
 (a)	Bends	4	·12	·06	·15	·10
		2	·15	·08	·17	·13
		1	·25	·15	·28	·20
		·75	·35	·22	·47	·33
		·5	·6	·4	1·3	·95
 (b)					Square	
					90°	135°
	Elbows	—	·85	·43	1·25	·65
 (c)	Bends with Deflector Vanes	$\frac{R}{D}$			90° 1 Vane	90° 2 Vanes
		1·5			·11	·15
		1·0			·13	·10
		·75			·16	·12
		·5			·70	·45
 (d)	Elbow with 'Duct Turns'	Single blade vanes				·35
		Streamlined double vanes				·10
 (e)	Tees or Twin Elbows	Take as for corresponding Bend or Elbow				
 (f)	Branch Inclined	α			V.P. in branch ×	
		15°			·09	Plus loss in bend
		30°			·17	
		45°			·22	
		60°			·44	
 (g)	Branch Radius Type. Circular or Square	$\frac{R}{D}$			V.P. in branch ×	
					90°	135°
		2			·2	·13
		1			·32	·22
		·75			·47	·34
		·5			·75	·55

FIG. 329.—*Continued.*

	Duct Element	α	V.P. in.	Factor
 (*h*)	Expander (Square or Circ.)	60° 30° 10°	Entrance „ „	·75 ·6 ·2
 (*i*)	Sudden Expansion	—	Entrance	1·0
 (*j*)	Contraction	60° 30° 10°	Exit „ „	·25 ·10 ·02
 (*k*)	Sudden Contraction	—	Exit	·5
—	Gratings: 50% free area	—	Free area	1·5
 (*l*)	Louvres: 90% free area 70% „ „	45° 45°	Free area „ „	·5 ·75
—	In General -	Abrupt changes in direction or velocity		1·0
		Easy changes in direction or velocity -		·5

(The above are quoted from Standard British and American Authorities.)

TABLE LXXXIV

METAL DUCT GAUGES AND WEIGHTS
(From *I.H.V.E. Guide*)

Specification A		Specification B	
Rectangular	Thickness S.W.G.	Rectangular	Thickness S.W.G.
Longer side up to 12 ins.	22	Longer side up to 12 ins.	24
Over 12 ins. „ 18 „	20	Over 12 ins. „ 18 „	22
„ 18 „ „ 36 „	18	„ 18 „ „ 30 „	20
„ 36 „ „ 54 „	16	„ 30 „ „ 48 „	18
„ 54 „	14	„ 48 „	16
Circular		Circular	
Up to 12 ins. diameter	22	Up to 18 ins. diameter	24
Over 12 ins. up to 20 ins.	20	Over 18 ins. up to 30 ins.	22
„ 20 „ „ 42 „	18	„ 30 „ „ 42 „	20
„ 42 „	16	„ 42 „ „ 72 „	18
		„ 72 „	16

WEIGHTS OF SHEET METAL, LBS. PER SQ. FOOT

Thickness of Sheet		Mild Steel Black	M.S. Galvanized after Manufacture. Black Plate of stated S.W.G.	M.S. Galvanized. Total Thickness of stated S.W.G.
S.W.G.	Ins.			
10	0·128	5·21	5·46	5·27
12	·104	4·24	4·49	4·3
14	·080	3·26	3·51	3·32
16	·064	2·61	2·86	2·67
18	·048	1·96	2·21	2·02
20	·036	1·47	1·72	1·53
22	·028	1·142	1·39	1·197
24	·022	0·897	1·147	0·952
26	·018	0·735	0·985	0·790

When estimating weight of sheet metal ducts 7–10 per cent. should be added for laps, rivets and scrap. Stiffeners, hangers and supports should be estimated separately.

POSITION OF THE FRESH AIR INTAKE

It might be thought that the purest supply of air would be obtainable at the roof of a building, but experience shows that in many instances this is not the case.

At roof-level air is certainly free from street dust, but chimneys of the same or neighbouring buildings may, with certain states of the wind, deliver their smoke or fumes right into the intake.

Two cases of this have occurred in the authors' experience. In both the offending source of contamination was due to a neighbouring building erected after the one in question. In one it was found that the main flue from the oil-fired boilers was brought up within a few feet of the intake.

In the other a kitchen incinerator flue, giving on occasion a variety of pungent effluvia, was found to exist some 30 or 40 feet away.

In the first case a solution had to be effected by a complete diversion of the intake shaft to another position at a lower level. In the second case a fan was installed to draw the incinerator smells away and discharge them at a great distance from the point of intake.

It would appear in most cases that the best point is to be found at lower levels, rather than high up. For instance, a case of an air supply drawn down from an open area adjacent to a busy main street is known to give ideal results. The dust drawn into the intake at such a point is mostly of a gritty nature, which can be easily extracted by a washer or air filter, and motor-car exhaust fumes do not appear to give trouble. A corresponding intake on the roof would draw in smoke and soot, both of which are most difficult, if not impossible, to filter.

If a point half-way up the elevation of a building can be found, this is probably the best. It must, however, be clear of windows where fire or smoke might occur, and particularly of lavatory windows.

There is no general solution to the problem of the fresh air intake position, as obviously every case requires examination of orientation, possible sources of contamination, position of air-conditioning plant, and so on.

METHODS OF CLEANING AIR

Wherever taken in, the air will contain dirt, particularly in cities, and must be cleaned to rid it of the dust, soot and other impurities. If the air is not treated, dirty marks quickly make their appearance on the ceilings or walls near every inlet grating; dust and dirt settle down on to papers, desks and furniture in the rooms, and the ducts become coated inside with a layer of fine black powdery material which has the unhappy knack of suddenly detaching itself in chunks, giving rise to serious complaints of damaged papers or soiled clothing. These effects are unfortunately noticeable to a less marked degree even when a so-called air filter is provided. Indeed, the cleaning of air to anything like 100 per cent. efficiency is one of the most difficult problems of air-conditioning.

It is important that the meaning of 'efficiency of filtration' should be properly understood. The percentage efficiency is obviously expressed by the ratio:

$$\frac{\text{Solid material extracted by filter} \times 100}{\text{Total solid material in entering air}}.$$

The question is, however, how the ratio shall be established, whether on a count basis, i.e. number of particles, on a weight basis, or on the 'blackening' basis.

If the number of particles is to be counted the usual method is to discharge a sample volume of air through a jet on to a gelatine-covered glass

slide, as in the Owen jet test. The slide is then examined through a microscope and the visible particles per unit area are counted. Those below a certain size, such as 0·5 microns,* are usually excluded as being too small for counting.

On the weight method screens of muslin of known weight are used. These are arranged so that air is passed through for a given period at a definite speed, one muslin being arranged after the filter under test and the other passing unfiltered air. The muslins are weighed again on very accurate balances on completion of the test. The muslins do not collect the whole of the dirt in the air and the weights of materials collected are only relative.

A series of tests on the latter method with two types of commercial filter was carried out at the instigation of the authors over a number of weeks. A photograph of the muslins is shown in Fig. 330, and the relative efficiency is shown under each. Each vertical row represents one complete week, the top one being from the unfiltered air, the middle one after a standard oil-coated filter, and the bottom row after a glass-silk filter. The percentages given under the two lower rows apply of course to the amount passing the filters (not by the muslin) and are obtained by deduction of the increase in weight of the muslins as compared with the unfiltered sample. The reason for the apparent decrease in efficiency from left to right is that the weeks were becoming increasingly foggy, as the test was carried out in November-December.

These tests, backed up by general experience, show that on a weight basis filters are much less efficient than on the counting basis, thus showing that the ultra-microscopic particles form a very large proportion of the total.

An oil-coated filter, for instance, may have an efficiency of 97 per cent. on the counting method, but the results by weight show something of the order of 70 per cent.

The count basis is not satisfactory, since, by omitting anything less than 0·5 micron a serious error is introduced. Obviously a 1 in. mesh would be 100 per cent. efficient if all particles less than the size of cricket balls were excluded.

In the blackening test a known volume of the uncleaned air is drawn through a filter paper by a suitable aspirator. A similar volume of cleaned air is at the same time drawn through a second filter paper. The relative blackening is estimated by reflected light using an electrical photometer. This test is more drastic from the point of view of filter efficiency than either of the other methods, since all particles whether large or small play their part in blackening. A filter giving 80 per cent. efficiency by weight test may give no more than 50 per cent. by blackening test.

So far as the building and its contents are concerned the blackening test is probably the best criterion, and if judged on this base present commercial filters show but poor results.

* 1 micron = ·001 mm.

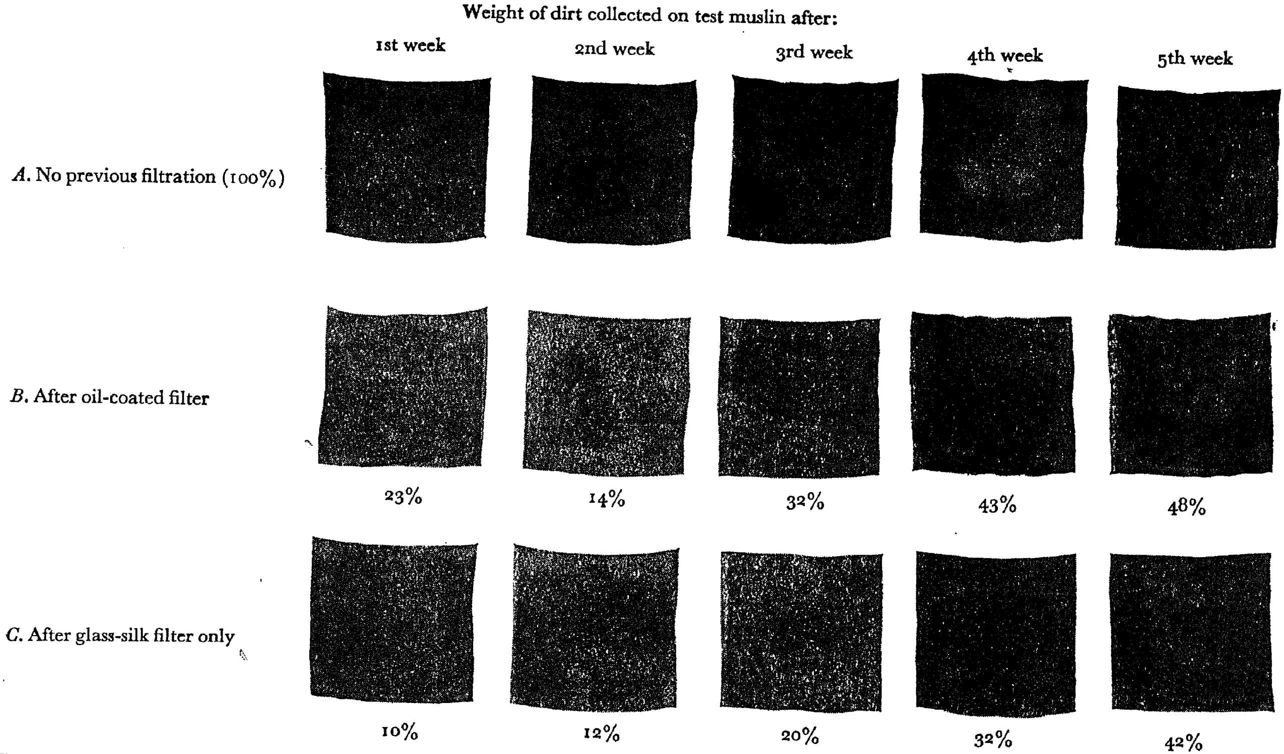


FIG. 330.—Results of Tests on Air Impurity by Weight Method. The five groups of three muslins show the dirt extracted from the three conditions of filtering (i.e. without filtering; after oil-coated filter; and after glass-silk filter). The outside conditions vary from clear to very foggy.

With filters having efficiencies of the order of 70 or even 80 per cent. as determined by any of these methods, it is apparent that a large amount of dirt passes through, as for every ten hours' run the equivalent of two or three hours' unfiltered dirt is delivered with the air.

The extremely fine particles referred to are probably due to the smoke or carbon in the atmosphere. Such material is not caught in a water spray air washer since (as is well known) water has a surface tension which repels particles of a greasy nature such as soot, and the two will not mix. Only the heavier and more gritty material can be brought down by a spray.

Oil-coated filters, as has been mentioned, do not collect this fine material, and recently there has been a return to fabric filters. An oil-coated filter followed by a fabric filter appears to give good results as the former removes the large particles and the latter is left only to extract the fine. As the fabric becomes dirty its resistance goes up and less air is passed, so that frequent cleaning or replacement of the material is necessary.

Oil-Coated Filters may be of the cell type shown in Fig. 331, the cells being often 18 in. square. These are cleaned by removing them in

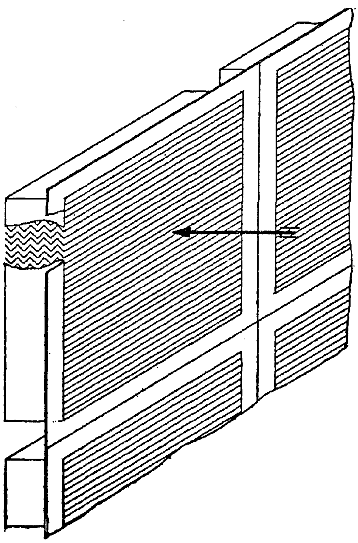


Fig. 331.—Oil-coated Filter, 'Ventex'.

sections, dipping in hot caustic soda solution, washing, re-oiling, draining and replacing. This periodical cleaning entails stopping the plant or having spare cells for use during the cleaning period. Self-cleaning filters have been developed to overcome this disadvantage, and one type is shown in Fig. 332. These are usually made in standard heights from

4 ft. to 11 ft., and the width made as required for the air volume in question. An air speed of about 400 ft. per minute on the face area is common with these types and the resistance is usually about $\frac{3}{8}$ in. to $\frac{1}{2}$ in. W.G.

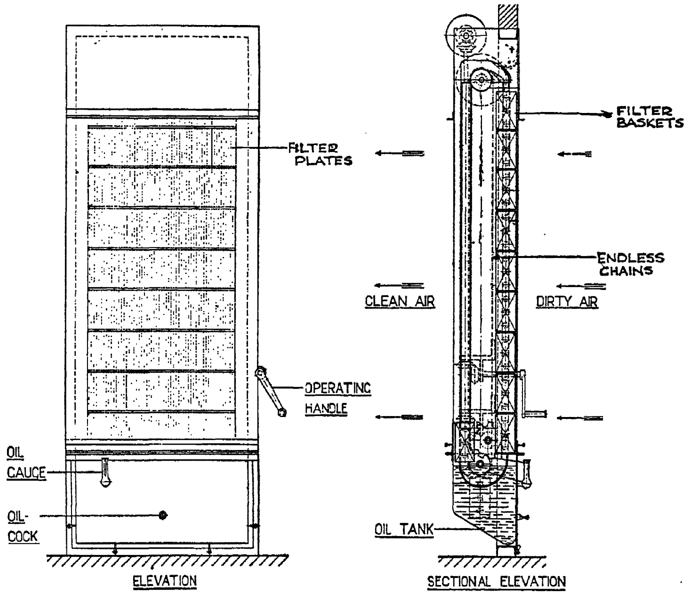


FIG. 332.—Self-cleaning Oil-coated Air Filter.

Fabric Filters—Types of fabric employed are blanket, serge, and cotton-wool about $\frac{3}{16}$ in. thick. A cellulose wadding or paper is also used in one type. The permissible velocity depends on the density of the filtering material and may be from 10 to 40 ft. per minute: the closer the weave and the lower the speed, the higher is the efficiency. Even with the latter rate it will be seen that the area required is very great compared with the face of an oil-coated screen, and for this reason fabric filters are usually arranged in a V-formation. One maker constructs the filter in cells 27 in. square, each to pass 1000 cu. ft. per minute, and the increased surface is obtained by a zig-zagging of the fabric within the cell itself. The resistance of such filters within the air speeds mentioned is about $\frac{1}{4}$ in. W.G. when clean, rising to $\frac{1}{2}$ in. W.G. when dirty. They then require replacement, otherwise the resistance will build up to 1 in. or over and the air flow be greatly reduced.

Partial cleaning by vacuum cleaner is possible with fabrics but not with the cotton-wool or paper types. When dirty the latter are thrown away and replaced by new. With very dirty air this may be required once a week; or, if placed after an oil-coated screen, once a fortnight or longer.

Another form of fabric filter is the Bag Filter, as Fig. 333, which works at a velocity of 10 ft. per minute through the material, which is a thick

'swansdown'. This may be cleaned by vacuum cleaner. Used continuously it will last up to six months. Unfortunately, though most efficient, this type of filter occupies considerable space. Its efficiency may be judged

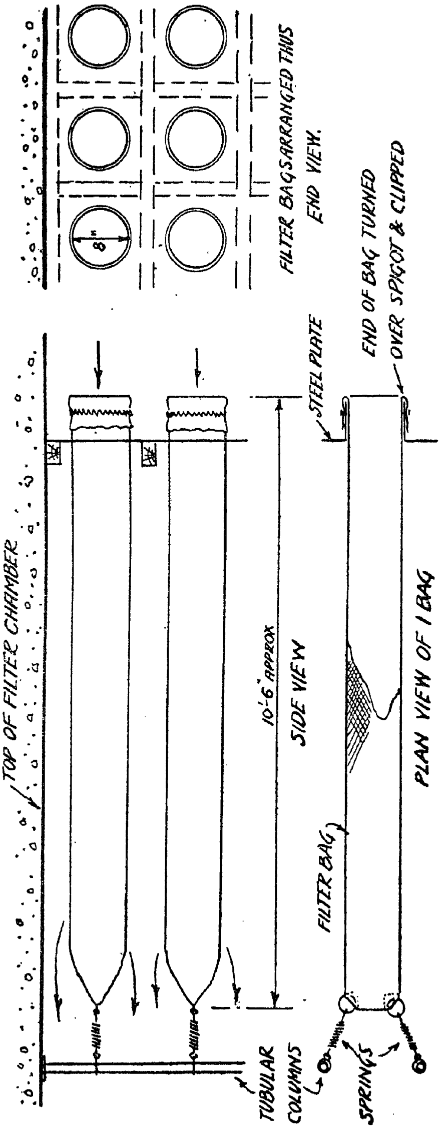


FIG. 333.—Filter Bag Type of Air Filter.

from the fact that it is found satisfactory in certain industries requiring the highest possible standards of air purity.

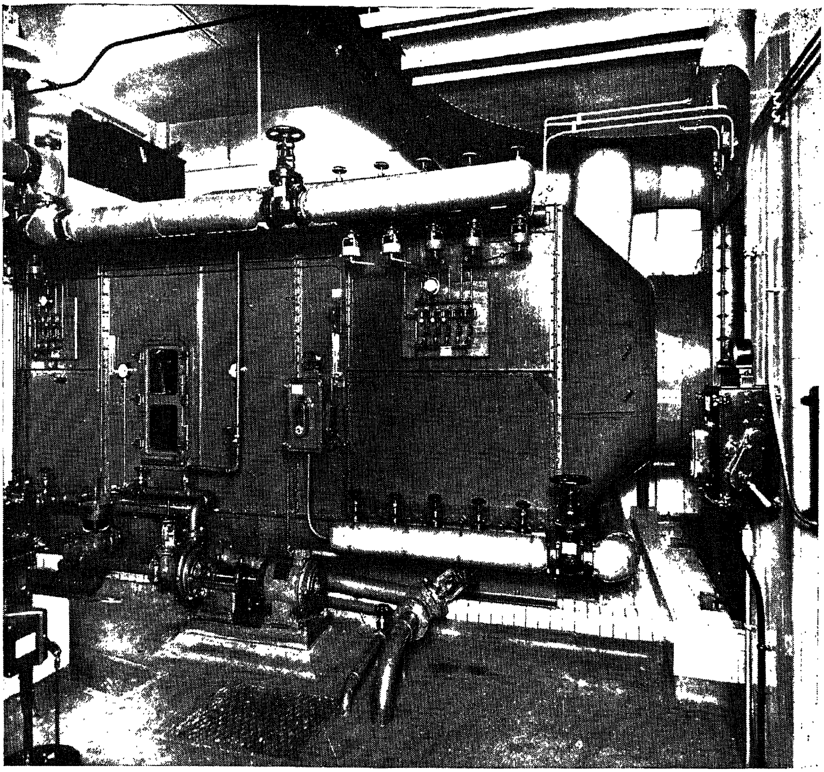


Plate XXV. Air-conditioning plant, showing air washer, pre-heater, main heater and fan (on the extreme right). The heating is by hot water. India House

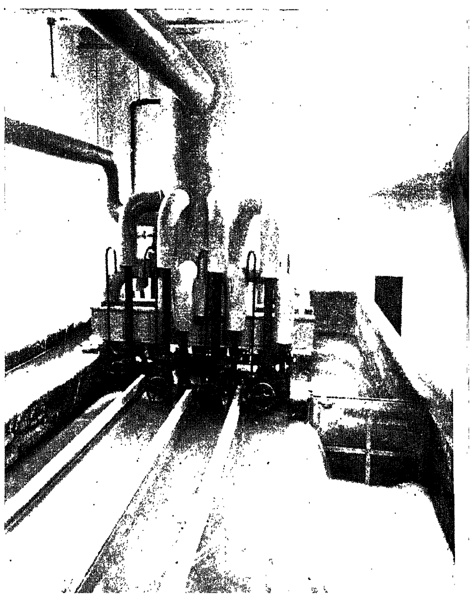
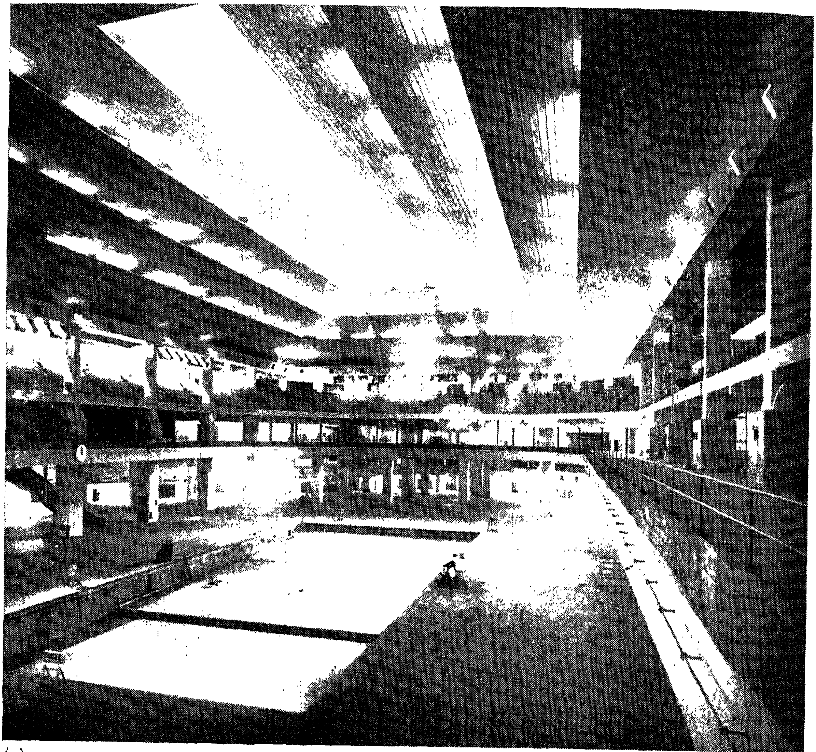
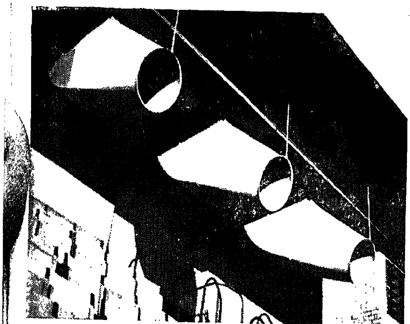


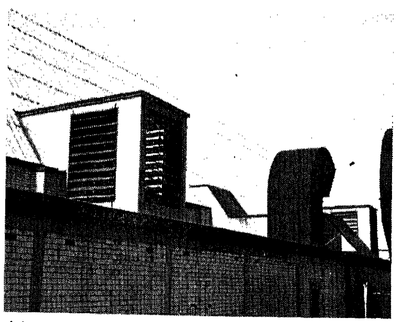
Plate XXVI. Evaporator for air-conditioning plant. Note cylindrical receiver, and cooled water flowing over weir to washer pump suction filter. Evaporator coils, submerged in water, are not visible



(a)



(b)

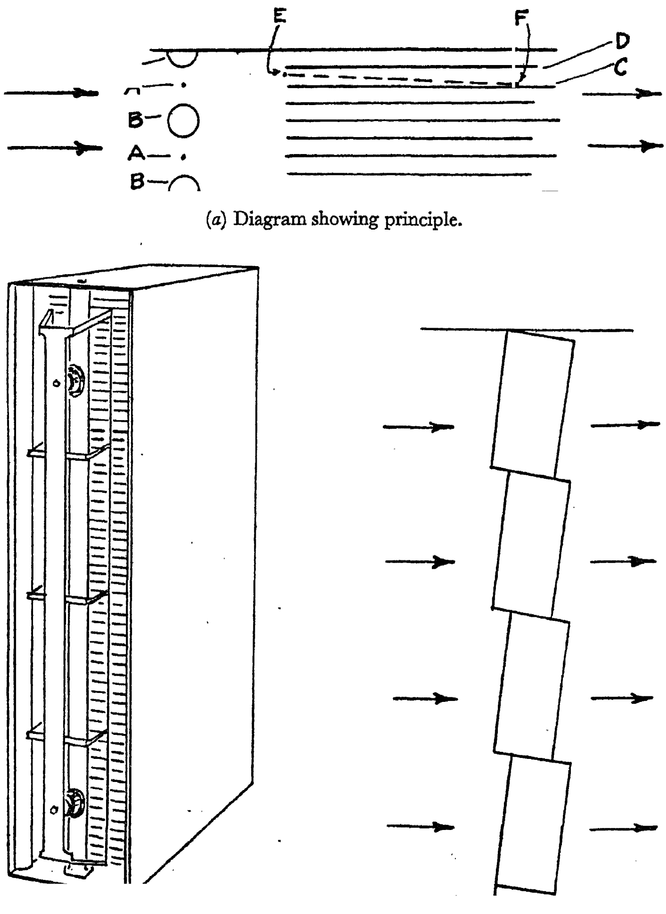


(c)

Plate XXVII. Earl's Court, London. (a) View of main hall showing nozzles at high level. (b) Detail view of nozzles. (c) Outside view of fresh air intakes and exhaust discharges

'Throw-Away' Filters—Filters of this type include filtering material of either glass-silk, steel, brass or aluminium wool, corrugated cardboard, or fibre. Their characteristics are similar to oil-coated filters, except that they are discarded and replaced when choked with dirt.

Electrical Precipitation—A new method of filtration of air promising



(*a*) Diagram showing principle.

(*b*) View of one cell.

(*c*) Arrangement of cells.

Fig. 334.—Electrostatic precipitator. The 'Precipitron' (Westinghouse).

good results is by electrical precipitation. This has been used for many years in power stations for cleaning flue gases, but has not until recently been developed (by Messrs. Westinghouse) for ventilation.

The principle is shown in Fig. 334. The filter is divided into cells, each

consisting of a series of fine ionising wires A between rods or tubes B, and followed by plates alternately earthed and charged, C and D.

The wires are charged at 13,000 volts positive D.C., and any particles of matter passing through the electrostatic field between these and the rods become similarly charged. The charged plates are at 6000 volts positive D.C. A particle arriving at E is already positively charged and is repelled by the positive plates, and eventually arrives at F on the earthed plate.

The earthed plates are oil-coated so that the dirt adheres, and are cleared by hosing down periodically. At twice-yearly intervals they are removed and re-oiled. As the cell becomes dirty the resistance does not increase as with other types of filter.

The electric supply is small, and is obtained from a high-voltage transformer and rectifier 'Power Pack'.

The efficiency is stated to be 90 per cent. on the blackening test when one cell measuring 36″ × 23″ × 8″ wide is passing 600 c.f.m. It is stated that it will remove all particles down to 0·2 micron and a good proportion down to 0·01 micron, which includes tobacco smoke particles. A small quantity of ozone is stated to be generated by the process, but this is not, of course, undesirable.

Unfortunately the cost is high, but it is to be hoped that future development will see it reduced, as it is believed to be the solution to one of the most troublesome problems of air treatment.

AIR WASHERS

An air washer, see Fig. 335, consists of a casing with tank formed in the base to contain water. Spray nozzles mounted on vertical pipes connected to a header deliver water in the form of a fine mist. The spray is projected either with or against the air current, or where two banks of sprays are provided, in both directions, one with and one against the air stream. The water for the sprays is delivered by a pump under pressure at 20 to 40 lbs. per sq. in., the water being drawn from the tank through a filter. The casing has an access door and internal illumination, and may be of galvanized steel, or constructed of brick or concrete asphalted inside.

As the mist-laden air is drawn through the chamber it requires to have the free moisture removed, and for this purpose eliminator plates of zig-zag formation are provided to arrest the water droplets. Some makes precede these with scrubber plates, down which water is caused to run by a spray pipe at the top in order to flush down dirt which has collected from the air.

Galvanized eliminator plates are liable to rapid deterioration. At the Bank of England, for example, they lasted about five years, and have been replaced with copper. Glass plates with serrated ribs have been used successfully by the authors elsewhere.

A single bank air washer will not saturate the air more than that corresponding to about 70 per cent. of the wet bulb depression. A double

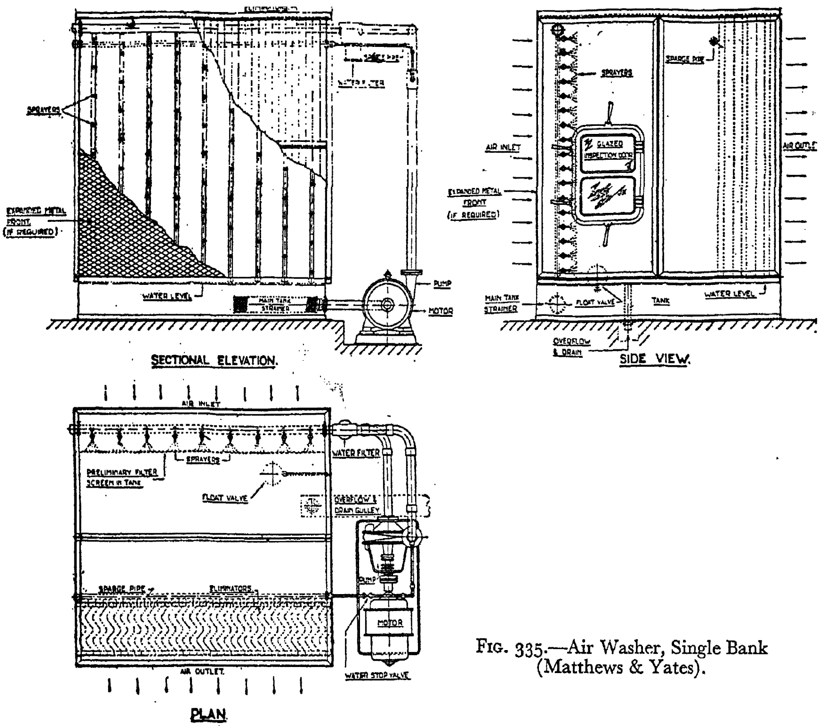


FIG. 335.—Air Washer, Single Bank (Matthews & Yates).

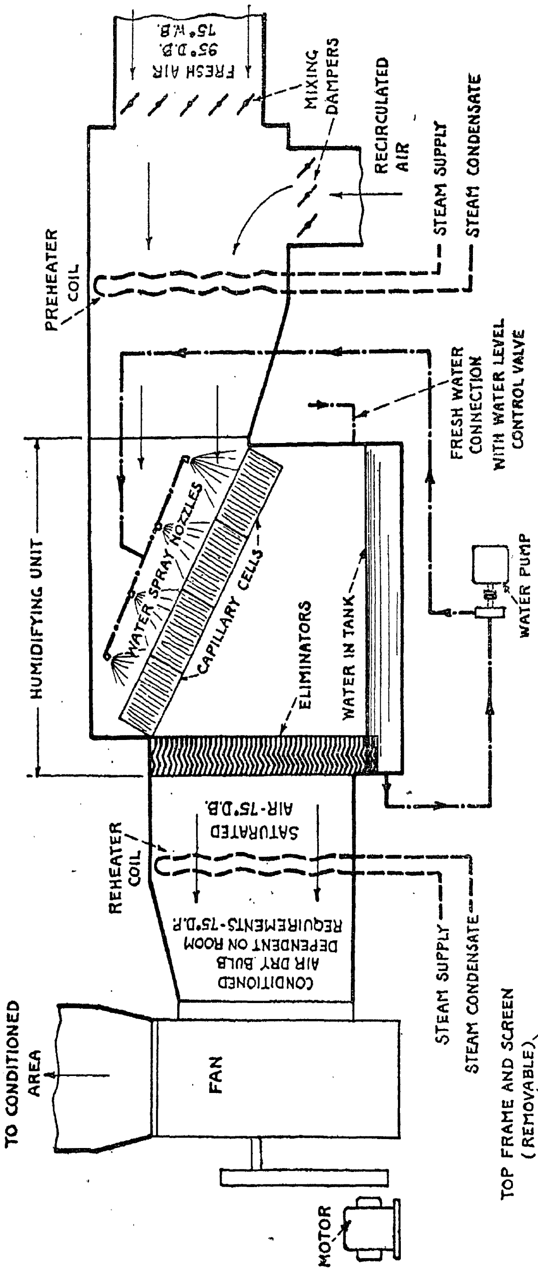
bank washer may reach 90 per cent. For full air-conditioning and dehumidification, where it is desired to saturate at the dew-point, a two bank washer is generally necessary with large spray nozzles to pass the necessary quantity of water for the temperature rise allowed. Spray nozzles vary in capacity from $\frac{1}{2}$ to 3 galls. per minute, and should be easily cleanable and of non-corrodible metal.

The speed of air through an air washer is usually 500–600 ft. per minute. The length is normally about 7′ 6″, but is increased with two or more banks of sprays necessary for cooling sometimes to as much as 12′ 6″. On the inlet side straightening vanes, or a perforated grille, are to be advised in order to distribute the air evenly over the whole area.

The tank is kept filled with a ball cock, with a hand valve for quick filling, and there is in addition a drain and overflow pipe. The pump is of normal type, preferably with insulated connections where noise may cause trouble, and arranged as to level so as to be flooded by the water in the tank, otherwise it will need priming.

Plate XXV shows the outside of an air-conditioning plant, and the washer, pump, main heater, and fan-suction will be clearly seen.

Capillary Cell Type Washer—This type, shown in Fig. 336, is a comparatively new development from the U.S.A. The air passes a series of cells



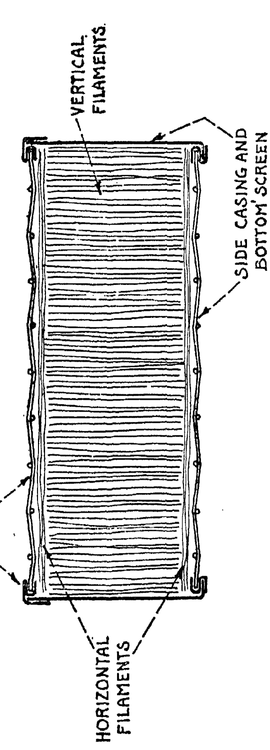


FIG. 336.—*Above*, Diagram of Conditioning Plant using Capillary Cell Washer (Air Refrigeration Corporation, New York; Messrs. Airscrew Ltd., London). *Below*, Detail of Construction of Capillary Cell.

inclined at an angle and containing glass fibres laid in straight formation to a depth of some 8 in. One cell, size 21″ × 21″, passes 1100 cu. ft. per min. Water at low pressure is caused to flow over the cells by flooding nozzles from a pump of low power consumption. The water and air have to negotiate the striations of the glass fibres together, and are thus intimately mixed so that nearly complete saturation is achieved. At the same time it is stated that the filtering efficiency is of a high order. The high saturation efficiency renders this type of washer particularly suitable for the evaporative cooling system, using large air volumes, referred to on p. 416. It may equally be used with refrigerated water for a full air-conditioning system, since as much as 10 galls. per min. per cell may be passed through; the normal rate is about 2 galls. per min.

For use as a filter when humidification is not desired, as in straight ventilation in summer, the pump may be started and stopped intermittently by a clock switch, so as to flush out dirt which has collected, but so as not to cause appreciable moisture addition.

Pumpless Washer—Yet another type of washer recently introduced, the Water Film Pumpless type, relies on the speed of the air to cause the water to spray. By means of baffle plates the air is caused to impinge on the surface of the water, thus causing violent turbulence. The air has to traverse this in the form of a series of bubbles, and it then has to pass an area of spray before exit. Dirt contained in the air is largely collected, due to the intimate mixture. The fan water gauge required is higher than usual as might be expected, and so far it appears to have been used mainly for industrial dust collection.

HEATERS

Heaters may be of cast iron, plain welded steel, finned steel, and finned copper. The four types are shown in Fig. 337. The latter are the highest in efficiency per unit of weight and size, though possibly difficult to clean if they become dirty. The plain tube type is almost self-cleaning since there is little on which the dirt can lodge, but occupies more space and is more costly.

Where placed after a washer, some means of protection of steel heaters is desirable, due to the moisture carried through. They are often zinc-coated.

Heaters are arranged in stacks or batteries, connected through a valve to the steam supply and having a condense trap on the outlet. Automatic controls may be to the whole battery or better to each stack separately, so that they operate in steps.

If the heater is warmed by hot water, larger headers are usually provided than with steam, and the outlet connects to the return main, generally with a lock shield regulating valve. Pump circulation is essential.

Heaters may be enclosed in sheet-steel casings, or built into brick-work surrounds with concrete slab top. This is often cheaper.

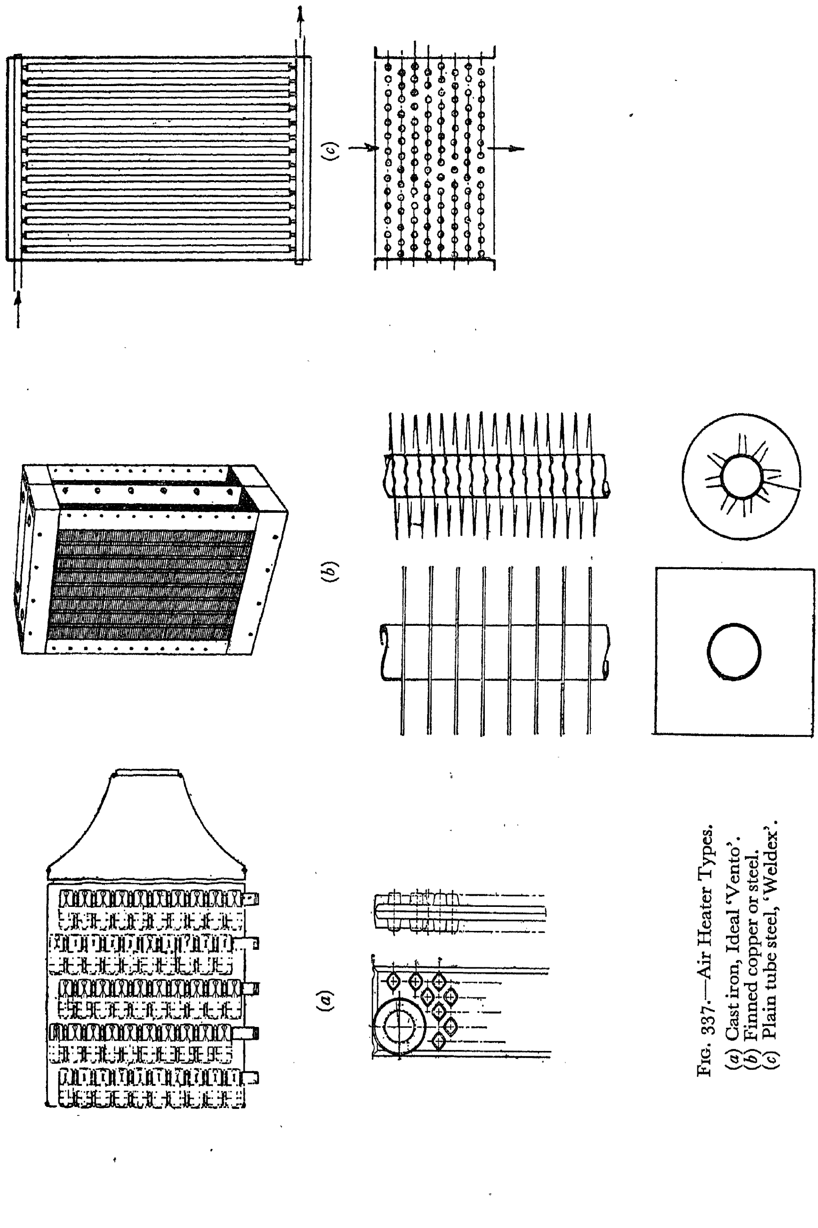


FIG. 337.—Air Heater Types.
(a) Cast iron, Ideal 'Vento'.
(b) Finned copper or steel.
(c) Plain tube steel, 'Weldex'.

The number of rows of tubes depends on temperature rise, temperature and nature of heating medium, i.e. whether steam or hot water, and air speed. The face area depends on air volume and free area between tubes. This is again determined by the velocity, and it is usual to fix this arbitrarily beforehand, generally between 800 and 1200 ft. per min. through free area. For sizing of heaters reference is necessary to makers' data.

TREATMENT OF AIR WITH OZONE, ETC.

Ozone is a gas with a chemical symbol O_3 (as compared with oxygen, O_2), and is stated to be present in the air in small quantities, particularly at the seaside, though recent medical opinion seems to contradict this.

It is unstable and readily breaks down into O_2 plus O. The detached atom appears to combine rapidly with almost any other available substance, thus producing a virulent oxidizing effect. This property gives ozone various applications, such as killing bacteria, eliminating smells and clarifying water and other substances. When ozone is mixed with air, organic material in the atmosphere is destroyed, and this is stated to have a beneficial effect on health. The strengths permissible in ordinary ventilation applications are probably too small to be effective.

Ozone by itself is probably odourless, but in practice it is inevitably mixed with various oxides of nitrogen which give it a rather penetrating smell unpleasant in concentration. For use in ventilation systems the mixture has to be extremely weak, about 1 to 5 parts per 100 million, when it is just discernible.

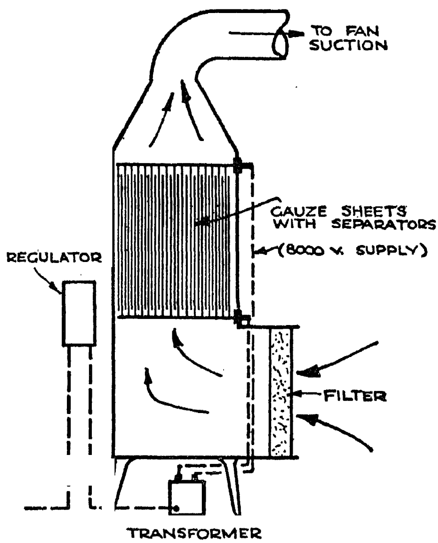


FIG. 338.—Section through Ozonizing Plant.

Ozone is produced by silent electric discharge or continuous sparking between parallel plates or sheets of gauze. An alternating current of about 8000 volts is applied to a series of such plates and the air is allowed to pass between *en route* to the fan. Fig. 338 gives a section through one type of ozone plant. A filter to remove dust is fixed on the intake to the ozone plant. Only a small proportion of the total air supply is drawn through.

Ozone is also produced in small quantities by the ultra-violet ray lamp, but this method has not yet been applied to ventilation.

Another method of treating air is by ionization. This is being experimented with in the United States, but though a mass of research has been done there is little practical result at present. Ionization may be described crudely as a change in the physical state of the air whereby the constituent atoms become electrically charged. This change is caused naturally by sunlight, and it is found that in dull dark weather the vitalizing ions are absent. It is supposed that by creating them artificially air equal to that on a bright summer day may be produced. We must wait some time before this is sufficiently developed for general use.

Mention should perhaps be made here of various disinfectants which may be introduced into the washer water tank to impart a faint pleasing smell to the air stream. These are no doubt beneficial for a crowded auditorium, wherein the effect is partly psychological.

REFRIGERATION FOR AIR-CONDITIONING

The cooling necessary for full air-conditioning is in nearly all cases effected by means of a refrigerating machine. Such a machine may be similar to the usual types of plant used for cold storage work, ice-making, etc., except that the temperature to be produced is not so low as in such applications.

For use with air-conditioning using a washer and dehumidifier, the water from the latter is returned to the evaporator or the refrigerating plant generally at between 45° to 50° F. depending on the dew-point to be maintained. In passing through the plant this is lowered from 6° to 8°. For the necessary heat transfer to take place, the refrigerant must be at some temperature below that of the water, but at the same time it must generally be slightly above freezing point to avoid icing of the tubes. Thus, in the example in Chapter XVIII, the following conditions obtained:

Apparatus dew-point	53°
Washer outlet	49°
,, inlet	43°
Water at evaporator outlet	$42\frac{1}{2}$°

The refrigerant in the evaporator would in this case be maintained at about 34°, giving $8\frac{1}{2}$° differential for heat transfer. This small temperature head means a very large cooling surface in the evaporator, and various devices have been developed to augment the heat transfer. Certain cases

exist where evaporators are run below 32°, making use of agitators, etc., to prevent freezing.

When a refrigerating machine is used in a direct-expansion air-conditioning system, the refrigerant is conveyed directly to the cooling coils in the airstream, and the surface temperature of these is dependent on the air-conditions required, on the form of coil surface and air speed over the coils. Again, refrigerant temperatures below freezing point are inadmissible owing to the risk of freezing on the surfaces when dehumidification is being performed, as such freezing would block the air flow.

REFRIGERATING MACHINES

The basic principle underlying all mechanically operated refrigerating machines is as follows:

A quantity of gas is highly compressed, which raises its temperature. The latent heat is then removed by condensing the gas to a liquid. Following this, the pressure is lowered and the liquid re-evaporates, extracting the necessary (latent) heat from any surrounding material, the temperature of which is thereby lowered. The refrigerant is now in its original gaseous state at low pressure, and the cycle is repeated.

A refrigeration plant (see Fig. 339), therefore, consists essentially of:

(*a*) A *compressor* to compress the refrigerating medium.

(*b*) A *condenser* to receive the compressed gas and liquefy it; the latent heat is taken out of the circuit by some external means. One method is to cool the condenser with circulating water, which may then be either run to waste or passed to a cooling tower for re-use.

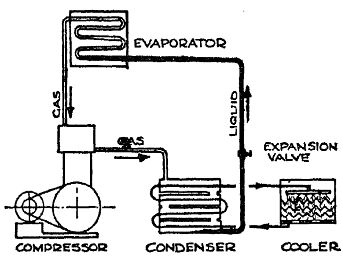


FIG. 339.—Diagram of Refrigeration Cycle.

(*c*) An *expansion valve* in which the pressure on the liquid medium is reduced.

(*d*) An *evaporator* in which the medium re-evaporates, extracting heat from the surrounding material, i.e. from the cooling water or air in an air-conditioning plant, or from the brine in a food refrigeration system.

Carnot Cycle—The ideal Carnot refrigeration cycle may be represented as in Fig. 340, in which heat is absorbed along the line AB, this being at the cooling or evaporating temperature T_1, and dissipated along the line CD by condensation at a higher temperature T_2.

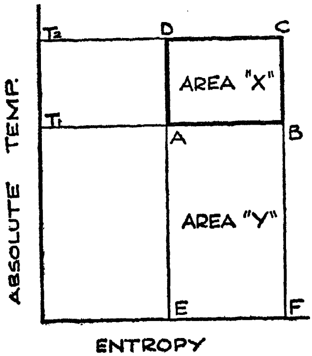


FIG. 340.—Theoretical Temperature-Entropy Diagram (Carnot Cycle).

The work done in compression of the gas (power input) is represented by the area X. The cooling effected is the area Y. The total heat to be disposed of in the condenser is represented by the area $X + Y$.

From this it will be seen that for a given power input X a much greater heat output is available at the elevated temperature T_2. This is the explanation of the Kelvin heat pump or reversed refrigeration cycle, which has often been toyed with for heating purposes as it appears to give efficiencies of anything up to 200 or 300 per cent. Except on a small scale, however, it has never really been put into serious use here.

The ideal Carnot cycle is not attainable in practice because of losses in compression, expansion, friction, leakage of valves, superheating of the gas on compression, etc.

From the diagram it will also be seen that the higher the refrigeration or evaporation temperature and the lower the condensing or cooling water temperature, the less work is necessary in producing a given refrigerating effect.

REFRIGERATING MEDIA

The factors affecting the choice of refrigerant will now be clear. A substance is required which can be liquefied at moderate pressures and which has a high latent heat of evaporation. The size of the compressor will then be kept down, and a relatively small amount of refrigerant need be circulated for a given amount of cooling. Such gases include ammonia, carbon dioxide, sulphur dioxide, and various organic gases.

Ammonia, whilst high in efficiency, and cheap, is ruled out in most cases for air-conditioning by the serious results which may follow a burst or leak.

CO_2 calls for higher power input for a given capacity, much higher pressures and requires a skilled engineer for its operation. Leakages are harmless but are very difficult to detect, the soap and water bubble test alone being possible.

Freon is an organic gas developed in the U.S.A. It operates at lower pressure than ammonia, is odourless, non-inflammable and non-toxic. For air-conditioning it is the most suitable refrigerant for piston type compressors.

SO_2 and methyl chloride are only suitable for small plants on account

of the large displacement necessary, and both are objectionable if a leakage occurs.

Air also may be used as a refrigerating medium. One method is to compress it to about 200 lb. per square inch and then allow it to expand through a valve. At this pressure the air is not, of course, liquefied, and the latent heat is not, therefore, available. The specific heat being small, very large volumes of air must be handled to achieve the desired amount of cooling.

Another method is to use pressures high enough to liquefy the air; at least 200 atmospheres are required. With either method large and expensive plant is necessary.

Water is also used as a refrigerant in a manner described later.

The refrigerants suitable for direct expansion into coils in the air-way are limited to CO_2 and Freon, the others all being objectionable, owing to their smell or inflammability, or inefficiency if conveyed through long pipes.

Table LXXXV gives a list of the more commonly used liquefiable refrigerating media, together with their characteristics and application.

TYPES OF REFRIGERATION PLANTS

In Table LXXXV the plants are placed in three groups, piston compression, centrifugal and steam jet.

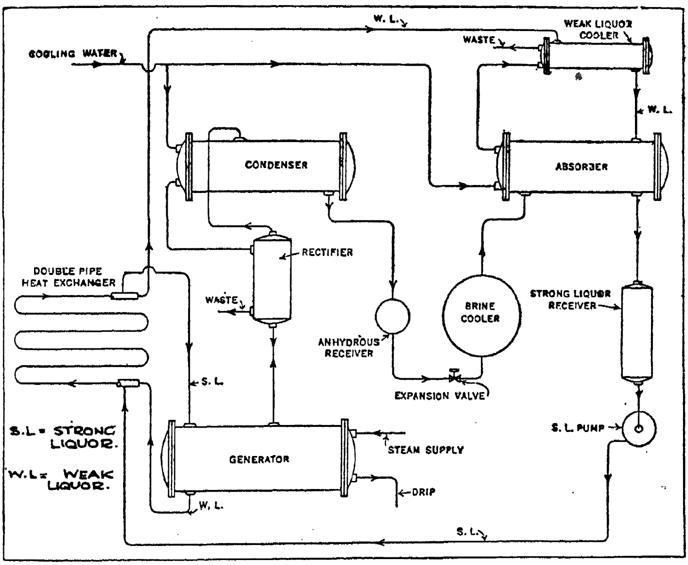


FIG. 341.—Diagram of Ammonia Absorption System.

Figs. 342, 343 and 344 show these three types diagrammatically. A fourth type, shown in Fig. 341, is known as the ammonia absorption system, but is not used for air-conditioning.

TABLE LXXXV—PROPERTIES OF REFRIGERANTS

Refrigerant	Symbol	Pressure in Lbs./Sq. In. Gauge		Boiling Point (Standard Pressure) °F.	Critical Temperature °F.	Volume of Vapour at 5° F. Cub. Ft./Lb.	Latent Heat of Evaporation at 5° F. B.T.U./Lb.	Theoretical H.P. per Ton Refrigeration	% Efficiency (Carnot Cycle = 100%)	Characteristics of Refrigerant	Used in
		Condenser 86° F.	Evaporator 5° F.								
Ammonia - -	NH_3	154·5	19·6	−28	271·2	8·15	565	0·993	83	Strong Irritant. Forms explosive mixture under certain conditions	Large Plants; Piston Compression Plants
Carbon-dioxide -	CO_2	1024	320	−108·4	87·8	0·267	115·3	1·75	47	Odourless. Innocuous. Does not support combustion	Large Plants; Piston Compression Plants
Dichloro-difluoro-methane ('Freon')	CCl_2F_2 (or 'F 12')	93·2	11·8	−21·5	222·7	1·48	69·5	1·050	81·5	Odourless. Non-toxic. Non-inflammable. Innocuous.	Large Plants; Small Plants; Piston Compression Plants
Sulphur-dioxide -	SO_2	51·7	−2·9	14	314·8	6·66	170·7	0·962	85·5	Strong suffocating smell. Toxic	Small Plants; Piston Compression Plants
Methyl Chloride	CH_3Cl	80·8	6·2	−10·7	289·6	4·53	178·5	1·025	82	Almost odourless. Anaesthetic. Inflammable	Small Plants; Piston Compression Plants
Dichloro-ethylene	$C_2H_2Cl_2$	−7·8	−13·9	122	470	63·0	136	0·985	83·5	As Freon	Centrifugal Plants
Water - -	H_2O	−14·08 (1·25" Hg)	−14·673 (0·057" Hg)	212	706	9880	1088	1·15	71·5		Steam Jet or Centrifugal Plants

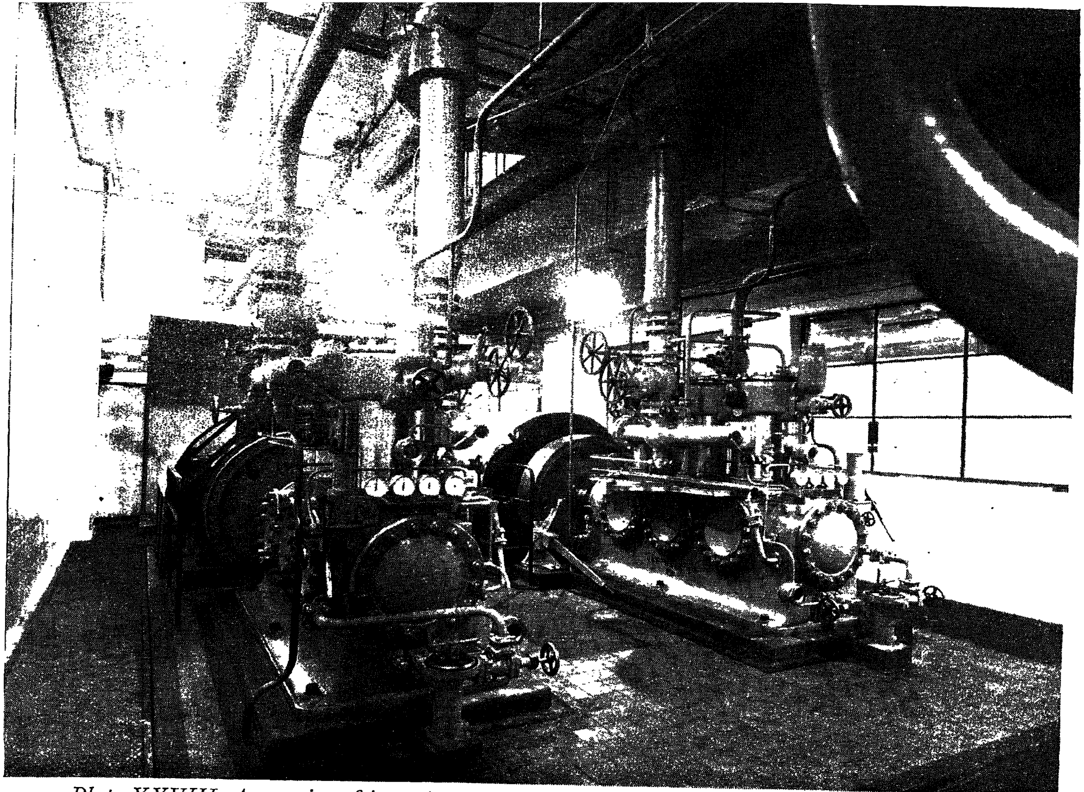


Plate XXVIII. Ammonia refrigerating compressors serving air-conditioning plant (see p. 545)

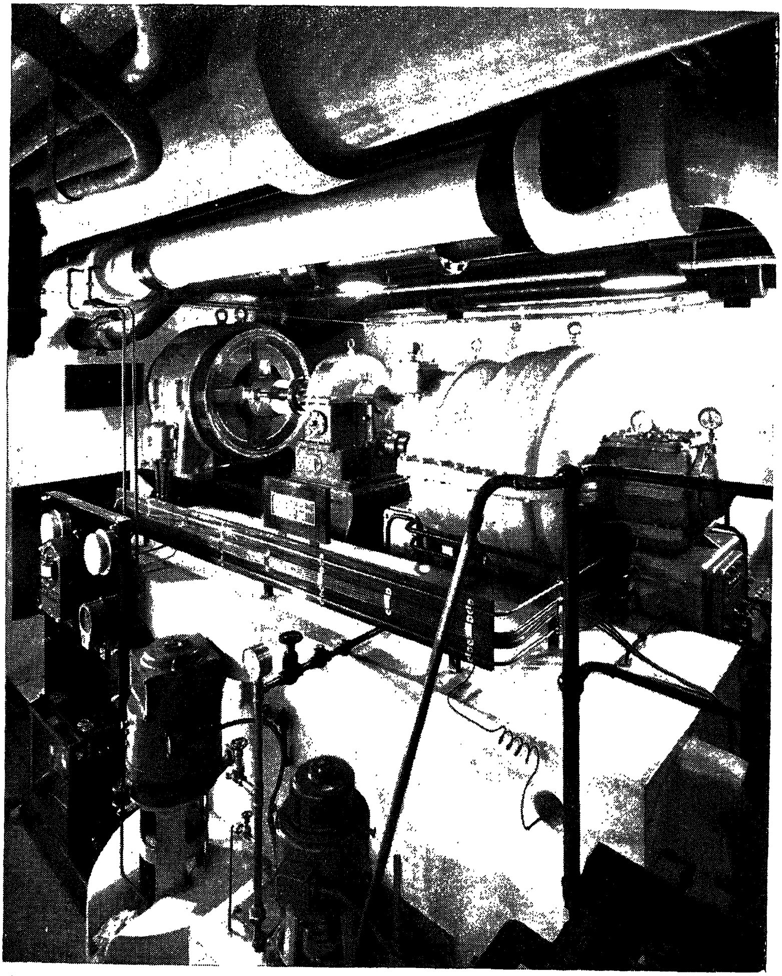


Plate XXIX. Centrifugal refrigerating compressor serving the air-conditioning plant in a City office building. (Carrier Engineering Co.)

Piston Compressors—The reciprocating type of machine may consist of one, two, three or four cylinders, according to the load, suction and discharge valves being operated by suction and pressure only and not mechanically.

Fig. 342 shows a diagram of a piston-compression unit, and Plate XXVIII illustrates two three-cylinder ammonia machines.

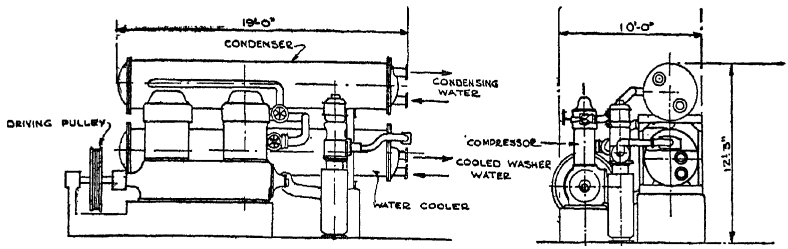


FIG. 342.—Piston Compression Unit, 200 Ton Capacity, using Freon.

In the single-acting ammonia, Freon, sulphur dioxide and methyl chloride machines, the crank case is filled with gas and forms the suction to the machine. The gland where the crank shaft passes through the casing is of rotary type and various systems are used for keeping this gas-tight. The cylinders are water-cooled. An oil separator is a necessity with ammonia machines to prevent oil from the crank case being carried over into the coils. Ammonia attacks many metals, including copper and all its alloys, which are not used in contact with the gas. Iron and steel alone are used.

Multi-cylinder machines are sometimes provided with a byepass, or unloading valve, on one or more cylinders, enabling the load to be varied either by hand or automatically. Variation of output may also be achieved by speed regulation.

With carbon dioxide the pressures are too high for the crank case to be used and the piston rod passes through a sliding type gland maintained gas-tight by oil under pressure from a pump. The portion under the piston is at suction pressure and the compression takes place above, unless of double-acting type. The cylinder is formed from a solid steel forging and is not water-cooled. CO_2 may be used in contact with any metal as it is inert except in the presence of oxygen.

Centrifugal Plant—The centrifugal compressor, one type of which is illustrated in Fig. 343 and in Plate XXIX, possesses many advantages for use with air-conditioning systems. Like a centrifugal pump it maintains a constant pressure head, and, therefore, a constant temperature head, almost regardless of load. Thus it automatically adjusts itself to varying cooling requirements, always maintaining a constant outlet temperature, whereas with the piston type compressor the temperature varies with the load, unless regulation in some form is resorted to.

The refrigerants, such as ethylene chloride (or dichlor-ethylene, sometimes called dielene), used with centrifugal compressors, are liquid at ordinary temperatures, are innocuous, transportable in drums instead of in high-pressure cylinders, and operate under vacuum in the plant so that leakages are inward rather than outward.

Being mechanically balanced, the compressor presents less of a problem in vibration isolation than reciprocating plants.

The general arrangement, as will be seen from the figure, is compact by reason of the mounting of the evaporator and condenser in one unit with

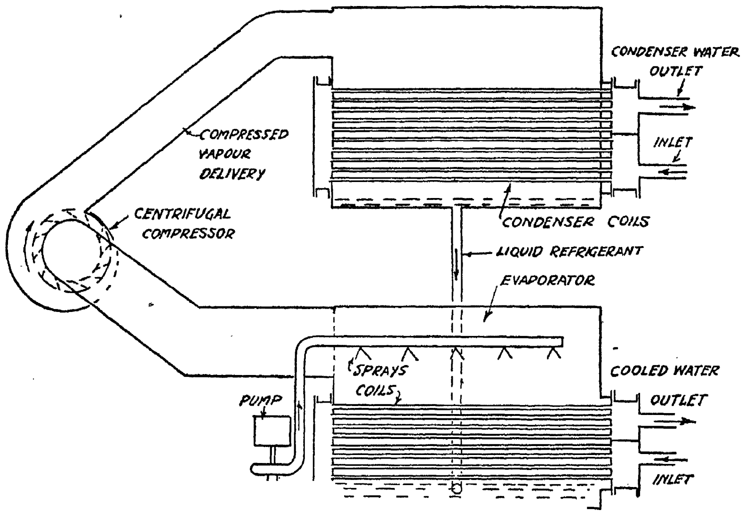


FIG. 343.—Diagram of Centrifugal Refrigerating System.

the compressor. Connecting pipes contain water only and losses and leakage are by this means materially reduced.

Steam-Jet Plant—An interesting type of refrigerator using water as the medium is that shown in Fig. 344. Its operation depends on the possibility of causing water to boil at low temperatures under high vacua. Thus, at 45° F., water boils under a barometric pressure of 0·30 inches of mercury, equivalent to a vacuum of 29·7 inches mercury.

The water used is the same as that circulated to the washer, so that all heat-exchanging surface on the evaporator side is avoided. The absence of any special refrigerant is an advantage and an economy.

The heat input with this equipment is much greater than with the positive compression types owing to the inefficiency of jet compression, and this calls for a greater amount of condensing water.

A variation of the same system, but using a centrifugal compressor in place of the steam-jet compressor, has also been developed. It avoids the above-mentioned disadvantage of high heat input. The great difficulty

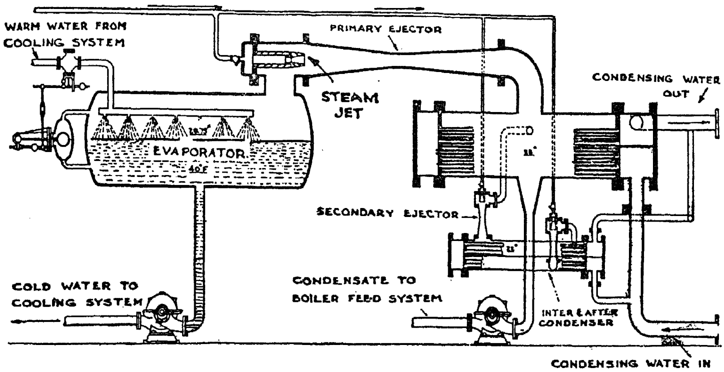


FIG. 344 (*a*).—Diagram of Steam-Jet Refrigeration Plant.

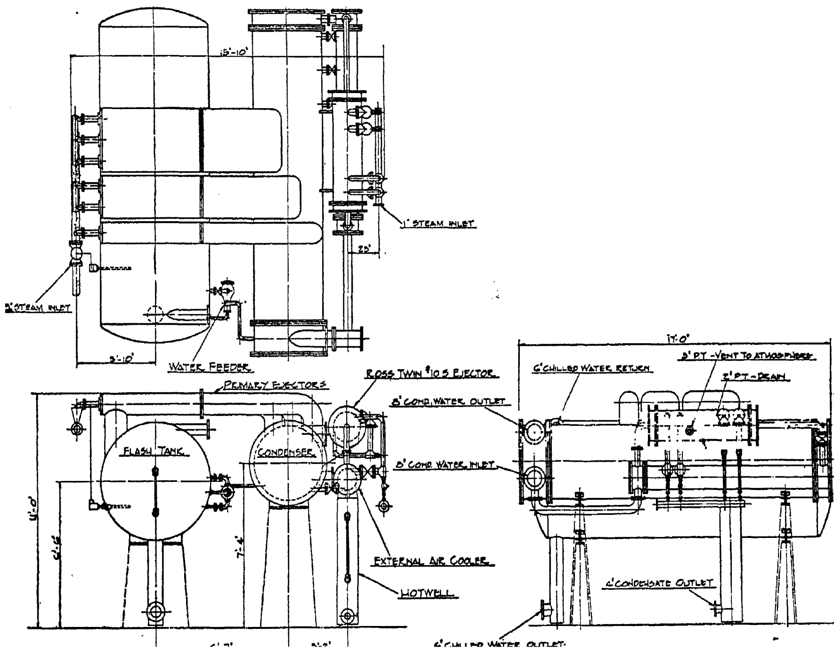


FIG. 344 (*b*).—General Arrangement of a 200-ton Steam-Jet Plant.

with both types is the maintenance of the extraordinarily high vacuum for long periods.

A centrifugal compressor using water vapour as the refrigerant has been developed, and is in use in a number of plants in U.S.A. and in a few in this country. Experience as yet is not sufficiently extensive to warrant further comment.

REFRIGERATING PLANT COMPONENTS

Evaporators—The evaporator is tubular in form, the tubes generally containing the refrigerant, the whole being immersed in the liquid to be cooled. In the *dry system* the evaporator coils are filled with vapour having little liquid present. When operated on the *flooded system* the liquid refrigerant discharges into a cylinder feeding the coils by gravity. As evaporation takes place the gas returns to the top of the cylinder and from there returns through the suction pipe to the compressor. An arrangement of this type is shown in Plate XXVI (facing p. 532): an agitator (not shown) maintains a rapid circulation of water over the coils, so as to achieve a high rate of heat transfer.

Another form of evaporator is the shell and tube type, but this is more suitable for cooling brine than for water, as the formation of ice outside the coils cannot be observed.

In the centrifugal machine the evaporator forms part of the unit, the water passing through the tubes and the refrigerant being dripped over the outside.

Condensers—The evaporative condenser, consisting of coils of gilled piping over a tank from which water is circulated and dripped over the pipes, is the simplest form of condenser. Its use is restricted to cases where the compressor can be near to the condenser, otherwise long lines of piping containing refrigerant under pressure are necessary.

More often in air-conditioning systems, the condenser takes the form of a heat exchanger of the double pipe, shell and tube, or multi-pass type. In each, use is made of the counter-flow principle so as to economize in pipe surface, the coolest water meeting the coolest refrigerant and the hottest water the hottest refrigerant, see Plate XXXI (facing p. 567).

Circulating piping from the condenser to the water cooler with these types alone traverses the building, and all equipment containing refrigerant is then confined to the plant room.

Evaporative Coolers—The heat extracted by the refrigerating machine, together with the heat equivalent of the power input to the compressor, raises the temperature of the condenser water by an amount depending on the quantity of water circulated through the condenser.

The lower the temperature of the condenser water the less power will be required to produce a given refrigerating effect, and conversely with a given size of plant the greater will be the amount of cooling possible.

Water from a well or from the main supply will always be the coldest,

the former at 55° F. and the latter at about 65° F. in summer. The quantity to be wasted, however, generally rules this method out. For instance, a 200-ton plant (2,400,000 B.T.U.'s per hour with power input equivalent to 400,000 B.T.U.'s/hour) with a 20° rise through the condenser would require

$$\frac{2{,}400{,}000+400{,}000}{20\times 10}=14{,}000 \text{ gals. per hour.}$$

This at 1s. per 1000 gallons would exceed the cost of current for running the compressor by about three times.

Cooling the water by evaporation is therefore generally resorted to. Evaporative coolers depend on the ability of water to evaporate freely when in a finely divided state, extracting the latent heat necessary for the process from the main body of water, which is then returned, cooled, to the condenser. In the case stated above the consumption of water with an evaporative cooler (assuming no loss of spray by windage) would be only

$$\frac{2{,}400{,}000+400{,}000}{10\times 1000 \text{ (latent heat approx.)}}=280 \text{ gals. per hour.}$$

The temperature of the cooling water is largely dependent on the wet-bulb temperature. An efficient cooler may bring the water down to within 4° or 5° of this temperature. Thus, with a maximum summer wet bulb of about 70° cooling water at 75° may be possible. This is often well below the dry-bulb temperature.

Evaporative coolers divide themselves into two categories, i.e.

Natural Draught.
Fan Draught.

The former is represented by:

(*a*) *The Spray Pond.* In this type the water to be cooled is discharged through sprays over a shallow pond in which the water is collected and returned to the plant. To prevent undue loss by windage the pond is usually surrounded by a louvred screen. Owing to the large area needed for the spray pond system, this type of equipment is seldom possible for air-conditioning applications.

(*b*) *Cooling Tower.* In this, the water is pumped to the top of a tower which contains a series of timber slats arranged so as to split up the water stream and present as large an area as possible to the air, which is drawn upwards due to the temperature difference, and by wind. The base of the tower is formed into a shallow tank to collect the water for return to the plant. Again, owing to its size and height, this type of cooler is not frequently used for air-conditioning.

(*c*) *Condenser Coil Type.* Where the refrigerating machine is near the

point where an outdoor cooler may be used, the condenser heat exchanger may be dispensed with and the refrigerant is then delivered to coils outside, over which water is dripped by a pump. The water collects in a tank at the base and is recirculated. A louvred screen is usually necessary surrounding the coils.

Fan draught systems are more commonly used owing to the compact space into which they may be fitted. Where possible they are placed on the roof, but if this is impracticable they may be used indoors, or in a basement, with ducting connections for suction and discharge to outside.

The following three methods are typical:

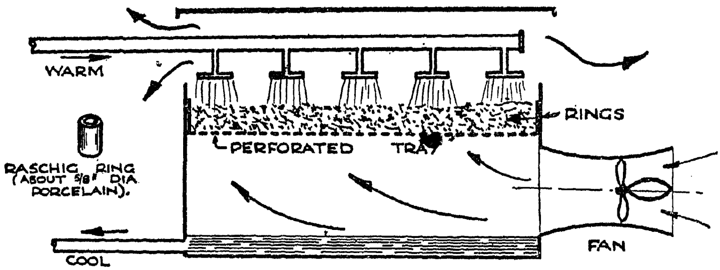


FIG. 345.—Raschig Ring Cooler for Condenser Water.

(*a*) *Raschig Ring Type*, Fig. 345. This employs a series of perforated trays containing porous unglazed porcelain rings giving a great area for contact between water and air. The water is dripped over the rings, and a fan is used to force the air through them.

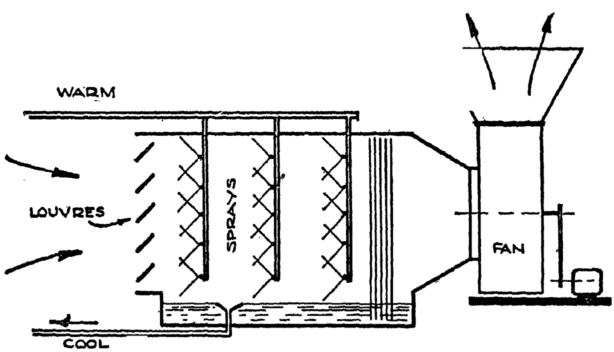


FIG. 346.—Fan Draught Cooler for Condenser Water.

(*b*) *Air Washer Type*, Fig. 346. This uses a normal type air washer containing several banks of sprays, base tank, and eliminators in the normal way.

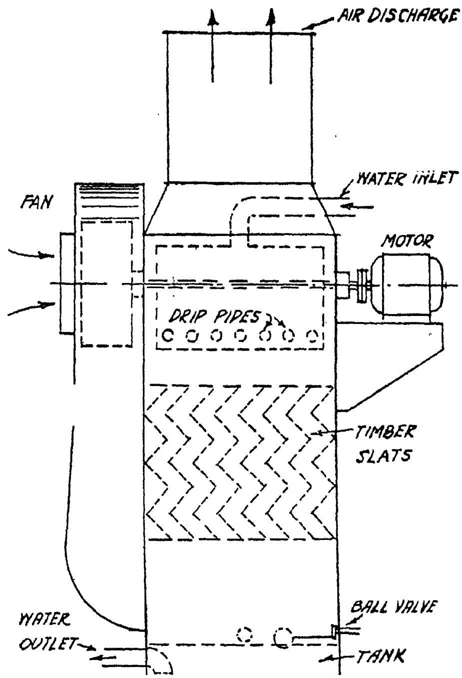


FIG. 347.—Forced Draught Cooling Tower (Heenan & Froude Ltd.).

(c) *Forced Draught Cooling Tower*, Fig. 347. In this the water is delivered to the top by the condenser pump, and delivers over timber slats as with the natural draught type. A cased fan is arranged to blow air upwards over the slats at high velocity so that a much reduced area of contact is required.

COOLING BY ICE

An alternative method of cooling and dehumidifying ventilation air is by means of ice.

The ice may be placed in blocks in the air stream, but it is more adaptable to control if crushed and placed in an insulated tank containing water. The water is cooled by the melting ice and may be used in a spray type air washer as with a refrigerated system; the mixing valve on the pump suction will regulate the amount of cooled water taken, so conserving the ice. The tank should have a capacity sufficient for a day's run, plus say 2 ft. water at commencement, and about 40 per cent. voids should be allowed in the ice, giving a capacity required of about 90 cu. ft. per ton of ice.

Ice has a latent heat of fusion of 144 B.T.U.'s per lb. and a sensible heat (from 32° to say 55°) of 23 B.T.U.'s per lb. Total 167 B.T.U.'s per lb.

The comparative cost of ice and refrigeration may be seen by taking the case given on p. 464 as an example:

Ice:	$\dfrac{685{,}000 \text{ B.T.U.'s per hour}}{167 \text{ B.T.U.'s per lb. ice}}$	=4,100 lbs. per hour.
	allow 6 hours per day	= 10·7 tons per day.
	cost, at 25s. per ton ice	=£13 7s. 6d. per day.
Refrigeration:	90 h.p., say 90 units per hour × 6 hours at 1d. per unit	=£2 5s. od. per day.
	Interest and depreciation on refrigerating plant over that of ice, assuming 40 days' use per annum at 10% on £4,000 $=\dfrac{400}{40}$	=£10 os. od. per day.

It will be seen that the refrigerating plant is more economical in running cost, and particularly so if the capital cost is ignored as being part of the building cost. The refrigerating machine is also more readily available when required, more convenient, and not subject to the uncertainty in the supply of ice which sometimes occurs in hot weather just at a time when needed most.

COOLING BY WELL WATER

Where a supply of water from a well is available in sufficient quantity and at a temperature of 50° to 55°, it may be used in an air-conditioning plant to give a certain amount of cooling. It will be appreciated that water at these temperatures does not permit of a lowering of the dew-point, hence no dehumidification is possible. Cooling by such means is therefore at the expense of humidity increase, and as has been pointed out before, the resulting comfort condition may not be improved thereby.

The water used in such a system must be run to waste unless a use for it can be found where impurities do not matter. Such cases are rare, hence the system has little application.

COST OF VENTILATION AND AIR-CONDITIONING INSTALLATIONS

The cost of the ventilating plant portion of the system, i.e. fans, ducts, heaters, washers, piping, etc., depends very largely on the lay-out, usage, occupancy, etc., and is not related to the cube as in the case of a heating system. Costs for buildings similar in size to the case considered may be misleading, and it is only possible to give widely divergent figures. The refrigeration portion is entirely dependent as regards size and cost on occupancy, fabric and sunheat gains.

Table LXXXVI gives figures for some examples carried out in recent years by the authors.

TABLE LXXXVI

COST OF VENTILATION AND AIR-CONDITIONING SYSTEMS

Building	Gross cube	Cost	Cost per ft. cube (pence)	Remarks
Private Office Building	1,050,000	£18,000	4·1	Complete air-conditioning in existing building
Bank Head Office (abroad)	2,900,000	£37,000	3·06	Complete air-conditioning
Hotel - - -	3,120,000	£35,600	2·74	,, ,, ,,
Government Office Building	1,127,000	£11,977	2·55	Complete ventilation and partial air-conditioning
Bank Head Office -	915,000	£8,900	2·35	Complete ventilation: air-conditioning to part basement
Bank Head Office -	3,130,000	£17,300	1·3	,, ,, ,,
Bank Head Office -	2,000,000	£10,355	1·24	Partial ventilation
Government Office Building	1,350,000	£5,620	1·0	Completely ventilated
Church - - -	933,000	£3,900	1·0	,, ,,
Residential Club -	978,000	£4,012	0·98	Partial ventilation
Factory - - -	350,000	£1,150	0·8	Plenum heating and ventilation
Town Hall - -	865,000	£1,730	0·48	Ventilation of Council Chamber and Committee Rooms
	Seats		Cost per seat	
Theatre - - -	1100	£2,850	£2 12s. od.	Complete ventilation
Theatre - - -	1150	£7,500	£6 10s. od.	Complete air-conditioning

* See note in Preface as to Costs.

CHAPTER XX

Combined Electrical Generating Stations

It is proposed to consider the case of a building or group of buildings for which electricity is generated by a private station, the waste heat from the generating engines being used for heating, etc.

Local and Central Generating Stations—It might be thought that in a civilized country, the normal procedure would be to receive cheap electricity from a public supply, and, generally speaking, this is the case in England to-day; but experience shows that there are exceptions to the general rule, and that in certain instances a local supply in the building has the balance of advantages, so justifying its installation.

The public supply should be cheaper for the following reasons:

(*a*) reduced cost of plant by the use of large stations where the peaks of one demand smooth out the valleys of another;

(*b*) reduced cost of running resulting from larger units being more efficient, fuel being cheaper when delivered and handled in large quantities, and the cheapest grade of fuel being usable.

Further advantages claimed for the public supply include:

(*c*) greater cleanliness of towns and smoke-abatement resulting from eliminating the private supplies;

(*d*) convenience of standardization, and synchronization;

(*e*) freedom from breakdown, arising from an inter-connecting grid which makes no single station indispensable.

Without suggesting that these advantages are not real, it is often found that any reductions in cost of plant and in running cost are discounted by:

(*f*) The reduced cost of generation is generally so small a proportion of the selling price that the consumer does not gain much from it.

Roughly the coal cost in a large station is about 0·1 of a penny, and the total generating cost less than 0·25d. per unit. Yet due to standing charges the consumer seldom pays, except in cases of thermal storage systems or large industrial consumers, less than ¾d. or 1d. a unit for combined heating, lighting and power (and often much more), so that a fractional saving on the 0·1d. component does not greatly affect the total cost.

(*g*) The increased cleanliness of our cities is very desirable, and might be achieved if the big generating stations were at the coalfields. But in practice, they are generally in close proximity to our cities, and the amount of coal burnt in our midst is therefore very little affected. The larger stations, it is true, go to great trouble to clean their flue gases, but the cost of cleaning must ultimately be paid for by the consumer.

It may be said that a properly run private station, with adequate stand-by plant, will give at least as good security against breakdown as a public supply. Further, the private supply is not subject to the risk of being extinguished by the faults of other people's systems, or by accidents in public streets, stations, or to overhead lines. Many private generating plants exist which have maintained an unbroken supply for thirty or forty years.

COMBINED GENERATION AND HEATING

The case for a combined generating station in a building depends largely on a technical consideration, namely, that in any heat engine only a small fraction of the fuel energy is converted into mechanical or electrical power, the rest (often 80 per cent.) being dissipated in various forms of waste, including loss of heat in boiler and stack gases, loss in engines and condensers, loss in generator and transmission system. In other words, of 100 units in the coal, only some 20 units are delivered to the consumer (see Fig. 173, p. 282).

In a combined system, i.e. a local station in a building combined with a heating installation, much of the waste heat can be utilized. Where steam engines are used, as much as 20 per cent. of the energy in the coal may be delivered as electricity and of the remaining 80 per cent., 30 to 40 per cent. can be recovered as useful heat. With oil fuel and Diesel engines, with heat recovery from exhaust gases and cylinder jackets, 40 to 50 per cent. of the fuel energy may be recovered as useful heat which is wasted where the station is not combined with a heating installation. Efficiencies as high as this are often balanced by increased cost of fuel and plant, but the point remains that the large public station makes no use of its surplus heat, while the private station can recover most of it.

With this advantage, added to the points already discussed, a case can often be made for the installation of a private combined station, on the score of cost, but the rates quoted for electricity from the public supply vary so much in different districts that in every case it is necessary to prepare an estimate comparing the total running costs with the two supplies, before any decision can be made.

With the combined system, the building acts as the cooling tower or heat absorbing unit to the engines (which in non-combined systems have to be provided with cooling towers, circulating water, etc., for this pur-

pose) and the engines act as the boilers or heating units for warming the building.

Steam Engines—A choice has to be made at the outset between what are in effect the only practical methods of electrical generation other than water power, i.e. steam, oil and gas engines.

Under steam engines come:

(*a*) reciprocating engines;
(*b*) turbines.

The reciprocating engine may be of lubricated type in which the exhaust steam is contaminated with oil, but it may be used in calorifiers or heating systems after passing through an oil separator. Even then it is not suitable for process work, laundries, or kitchens, where the slightest trace of oil would be objectionable. Alternatively, they may be non-lubricated, in which case care has to be exercised in the selection of the metals for piston-rings, slide-valves, etc., but provided suitable precautions are taken, the steam is useable for all purposes.

The reciprocating type is particularly suitable for smaller-sized installations, as it remains economical in steam consumption at comparatively low pressures and sizes, and is unaffected by variation in the back-pressure of the exhaust. Engines of this type are sometimes called upon to run 'condensing', i.e. under vacuum, at certain periods, and at others under a back pressure up to 20 lbs. per sq. inch or more.

Turbines are most economical for large installations, with high initial steam pressure, and where the back pressure is at atmospheric or as much below as possible. It must be remembered that exhaust steam at much below atmospheric pressure is useless for heating purposes, owing to its low temperature.

They are not suitable where considerable variation in back pressure is likely to occur, owing to the varying temperature causing expansion and contraction strains in the blades and casing. A common method of using turbines is on the system known as 'bleeding', by which a proportion of steam is exhausted for process or heating work from one of the intermediate stages, to give such pressure as is required, e.g. 30–50 lbs. per sq. inch. The remainder passes on through the low pressure stages and is exhausted under vacuum to a condenser.

The exhaust steam from turbines is uncontaminated with oil and may be used for all purposes.

Superheated steam may be used in a lubricated reciprocating engine or in a turbine, higher economy being achieved thereby. In the smaller types of plant saturated steam is usually adopted for simplicity, and provided there is ample demand for exhaust steam, and there are few occasions on which exhaust steam is blown to waste, there is little point in striving for extreme economy in steam consumption, since such steam as is not supplied by the engine has in any event to be made up to the heating system through

a reducing-valve direct from the boilers. In the case of turbines superheated steam gives drier conditions at the bottom end, hence less blade erosion.

TABLE LXXXVII

APPROXIMATE B.H.P. OBTAINABLE FROM 10,000 LBS. OF DRY STEAM PER HOUR FOR VARIOUS INLET AND EXHAUST PRESSURES

Inlet Pressure (Lbs. per sq. inch Gauge)	Exhaust Pressure															
	Inches Hg.				Lbs. per sq. inch gauge											
	20" Vac.	15" Vav.	10" Vac.	5" Vac.	0	5	10	20	30	40	50	60	70	80	100	125
40	235	225	215	205	200	—	—	—	—	—	—	—	—	—	—	—
50	275	265	250	240	235	200	—	—	—	—	—	—	—	—	—	—
60	300	290	280	270	260	225	190	—	—	—	—	—	—	—	—	—
80	375	360	345	330	320	275	240	185	—	—	—	—	—	—	—	—
100	440	425	410	395	380	330	280	225	175	130	—	—	—	—	—	—
125	485	470	450	435	420	380	330	260	215	175	140	—	—	—	—	—
150	505	490	470	455	440	410	370	310	250	210	170	145	125	—	—	—
175	530	510	490	475	460	430	400	340	280	235	205	180	150	130	—	—
200	540	520	500	485	470	445	420	365	320	275	235	205	180	155	120	—
225	550	530	510	495	480	455	430	380	340	305	265	225	205	180	140	105
250	—	—	—	—	490	465	440	390	360	325	285	250	225	200	160	120
300	—	—	—	—	—	—	460	420	385	355	325	285	255	225	190	155
350	—	—	—	—	—	—	480	440	405	375	345	310	275	250	215	180
400	—	—	—	—	—	—	—	460	430	400	370	330	300	270	235	200

Power Increase for Superheated Steam: 17½% for 100° Superheat; 25% for 150° Superheat. Figures are for one unit taking 10,000 lbs./hour. For two at 5000 lbs. reduce output by 5%. For larger units than 10,000 lbs. better efficiencies possible. This table is only applicable to reciprocating engines. (Table reproduced by kind permission of Messrs. Belliss & Morcom, Ltd.)

As a rough guide to the power obtainable from 10,000 lbs. of steam used in a reciprocating engine under various conditions of inlet and exhaust pressure, Table LXXXVII above may be useful. To take an example:

Inlet - - -	200 lbs. per sq. in.
Exhaust - - -	20 ,, ,,
B.H.P. available	=365

If the exhaust is at atmospheric pressure the b.h.p. goes up to 470.

Piping Arrangement in Steam Plants—Fig. 348 indicates diagrammatically the arrangement of a combined (steam) plant. Steam from the boilers is delivered at high pressure to the engines through the pipes marked in heavy black lines, and to the auxiliaries and turbine-driven circulating pumps for the heating and hot- and cold-water systems. Exhaust steam from these units is collected into an exhaust steam manifold, from which are supplied the heating and hot-water calorifiers and the outgoing steam

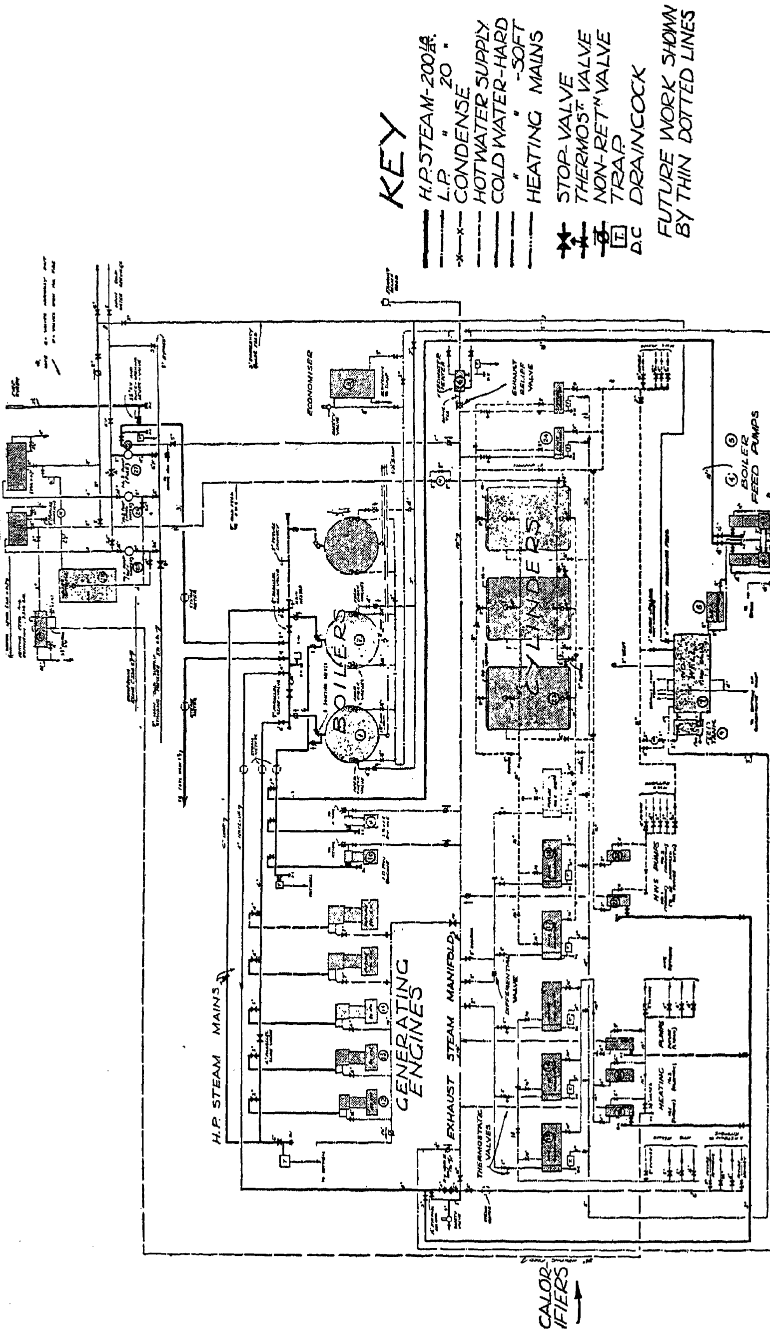


Fig. 348.—Diagram of Piping for Combined Generating Station.

mains serving kitchens, laundry, etc. In order to maintain the requisite pressure in the exhaust steam manifold at times when there is an insufficiency of waste steam available, a reducing valve is arranged from the high-pressure steam line so as to make up whatever is required in addition. When more exhaust steam is available than is called for by the calorifier and other services, an exhaust relief valve at the end allows the surplus to pass to atmosphere.

The diagram also indicates the arrangement of the boiler feed, in which all the available condensation is brought back to a hot well kept at a minimum level by means of a feed tank containing a ball valve supplied from the hot-water system. The feed is then metered before passing to the boiler feed pumps, and thence through the economizer to the boilers.

The heating systems are circulated by means of pumps, one electric and one turbine (an additional future turbine pump being provided in this case).

The temperature of the outgoing water is controlled by means of thermostatic valves in the steam supply to the calorifiers.

The system is arranged with the calorifiers for the hot-water supply system connected to storage vessels of unusually large capacity so as to be able to take exhaust steam over the 24 hours. The hot-water supply calorifiers are not controlled thermostatically, as their heating capacity is limited and they will always be able to take the steam supplied to them without unduly raising the temperature of the large storage vessels and great extent of mains. In order to ensure that the outgoing hot-water supply is at the requisite temperature, boosters are provided as indicated.

Diesel Engines—The term Diesel engine is now understood to mean any type of internal combustion engine using oil fuel on the compression-ignition principle. The various types are designed to run on any of the grades of oil now available. Oils suitable for burning in boilers are in some cases suitable also for Diesels, though for the smaller engines special Diesel oils are produced.

In the true Diesel an air compressor is part of the engine. Air is introduced to the cylinder on the suction stroke, to be compressed on the compression stroke near the top of which the oil is introduced with a blast of highly compressed air through a nozzle or atomizer. The temperature of the air under compression being above the ignition temperature of the oil, a flame is propagated, causing rapid expansion or explosion, this producing the power stroke. At the end of the power stroke the exhaust valve opens, and the following upward stroke of the piston discharges the exhaust to atmosphere. The suction stroke then follows and the cycle repeats. This is a four-stroke engine, though two-stroke Diesels have also been developed.

Owing to the troubles and complications associated with high pressure air compressors, there has been developed the 'Airless Injection' engine. This draws atmospheric air into the cylinder on the downward or suction

stroke to be compressed on the upward or compression stroke; oil without an air blast is introduced just prior to top dead centre and the power stroke and exhaust stroke follow. This is a four-stroke cycle, it is cold starting, i.e. does not require outward application of heat before starting, and is comparatively simple to operate and maintain. It has become by far the most common type for general purposes.

There are other types such as two-stroke hot bulb in which a blow lamp is used for warming up before starting, and the two-stroke airless injection cold starting. Speeds tend to increase; 200 r.p.m. was at one time usual, now 400 and 600 r.p.m. are quite common, whilst for traction work 1000 r.p.m. and over are general.

In all types circulating water is passed through the cylinder jackets for cooling, and the temperature of this should not be allowed to go above 120° or 130° F. Thus it is only of use for radiator heating where designed for low temperatures, but may be used with any panel system or as a feed to a domestic supply.

The exhaust gases may be passed through a waste heat boiler for the heating of water for radiators or panel system, or for hot-water or steam production.

Where steam is not required for other purposes, and where the summer demand for waste heat is small, the Diesel engine offers advantages as

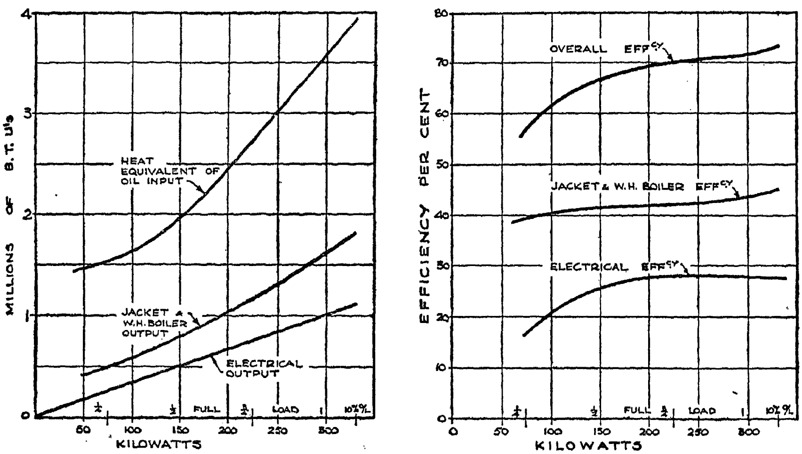


FIG. 349.—Results of Tests on 440 h.p. 300 kw. Diesel Generating Set.

compared with steam on account of its higher thermal efficiency and smaller light load losses.

The power output at ¾ to full load is equivalent to about 30 per cent. of the heat input of the oil, and the waste heat extracted from the engine jackets is about 30 per cent. and from the exhaust gas waste heat boiler about 10–20 per cent. This gives an overall efficiency of 70 to 80 per cent.,

the remaining 20 to 30 per cent. being lost in radiation, friction and residual heat in flue gases. Fig. 349 gives the results of a series of tests on some 300 kw. Diesel sets, and shows how the heat input and output and efficiency vary at different loads.

The fuel consumption of these engines, typical of many, is as follows:

¼ load	-	·98 lbs. oil per kw.	Full load	-	·595 lbs. oil per kw.
½ load	-	·675 lbs. oil per kw.	10% overload		·655 lbs. oil per kw.
¾ load	-	·60 lbs. oil per kw.			

Steam and Diesels compared—Comparing the above figures with a steam engine of the smaller sizes common to the type of installation under consideration, the power output would be equivalent to about 10 per cent. of the heat input to the boilers, and the exhaust steam available for heat recovery about 40 to 50 per cent., giving an overall efficiency of 50 to 60 per cent.

It will be seen from this that the heat to be wasted when there is little demand for it, such as occurs in summer with a combined system, is 20 per cent. more in the case of a steam plant than in the case of Diesels.

Other points to be borne in mind when comparing steam and Diesel plants are:

(*a*) Space occupied by Diesels is greater than steam engines of equal size, but less than steam engines plus boilers and auxiliaries.

(*b*) Oil is more conveniently handled and stored than coal.

(*c*) Maintenance costs of Diesel engines are heavier than with steam engines. Lubricating oil consumption is heavier.

(*d*) Initial cost of steam engines and generators is of the order of £6 to £8 per kilowatt, but the cost of boilers and auxiliaries may bring this up to £10 or over unless the boilers are common to any system, as is often the case. Initial cost of Diesels and generators excluding tanks and auxiliaries is generally about £10 per kilowatt. An inclusive figure may be £12 to £15 per kilowatt installed.

There are some conditions where it is advantageous to adopt a steam plant for base load with a Diesel set for use at peaks and night-time, etc.

Gas Engines—Gas engines are very similar to Diesel engines in regard to first cost, size, efficiencies and losses. The cylinders have to be slightly larger for a given horse-power, and electric ignition provided.

With Diesel oil at 80s. a ton, oil costs 2·4d. per therm. Hence gas needs to be supplied at about the same figure to be competitive. Usually the rates are higher, which accounts for the general preference for oil.

A recent development is the *gas-cum-oil* system in which the normal Diesel engine is used with gas as the main fuel, the oil serving only for ignition. In this way the oil consumption is reduced to about 10 per cent.

of full load oil, but the proportion may be varied whilst the engine is running from full-gas to full-oil. This makes a very flexible arrangement, as either fuel can be used at will according to availability and cost. Electric ignition is not required. The main engines in Plate I operate on this method.

General—The case of hospitals and institutions in general probably shows the private generating plant to best advantage, since in these the heat requirements are fairly steady, and even in summer there is always some demand for waste heat. Many cases exist, however, where it is found that exhaust steam has to be blown to waste or discharged to a condenser, where in turn the heat is lost to the circulating water. Such loss has to be allowed for in the heat balance sheet, and obviously adds to the cost of current generated privately. The simplest method of determining this loss is to construct load curves both for the electrical and steam requirements. From these it can be seen at what times of the day and in what quantity exhaust steam is wasted. In cases where this loss is considerable, it is usually found desirable to provide one or more Diesel engines for running during such periods.

Running Costs of Combined Stations—The running costs may be divided into two parts, one for the heating system, etc., and the other for the electrical generation.

The former is an orthodox arrangement with low pressure steam either used direct or in calorifiers, and also supplying kitchen, laundry, etc. The running cost may be arrived at on the lines indicated in Chap. XVI, taking the cost of the plant for depreciation purposes as that necessary to provide low pressure steam without combined generation. It is outside the present comparison.

Considering the electrical part of the plant, the running cost must include the following items:

(*a*) Fuel. This may be credited with the amount of fuel required to provide the heat required.

(*b*) Interest and Depreciation of plant and buildings. This will include the engines, generators, switchboard and all purely electrical plant, but only the *extra* cost of boilers, etc., over those required for a separate heating and hot-water supply system.

(*c*) Lubricating Oil and Stores.

(*d*) Insurance.

(*e*) Repairs and Maintenance.

(*f*) Labour and Management.

Actual examples of stations with steam and Diesel plants are discussed below.

EXAMPLE OF RUNNING COST CALCULATION: STEAM SYSTEM

Type of Building - - - - -	Hospital
Number of Patients - - - - -	1150
Number of Staff - - - - -	187
Total - - - - -	1337

Electrical Load and Consumption—

	Installed Load kw.	Maximum Demand kw.	Units per annum
Lighting - - - -	210	150	120,000
Power - - - -	35	24	140,000
Laundry - - - -	26	22	30,000
Electro-Medical - - -	35	20	8,000
Cooking - - - -	321	150	200,000
Sundries - - - -	15	10	13,000
Pumping - - - -	20	15	20,000
Total - - -	662	391	531,000

Estimated overall diversity factor, 0·7 × total max. demand.
Estimated overall maximum demand, 391 × 0·7 = 280 kw. approx.

Total units (531,000) is estimated to be divided as follows:

Winter - - - - - - -	350,000
Summer - - - - - - -	181,000

Engines Installed—

3 at 150 kw. - - -	= 450 kw.
1 at 50 kw. - - -	= 50 kw.
	500 kw.

(Allowing one engine under repair, maximum available = 350 kw.)

In this case no battery is provided and the small set is intended for night running.

Steam pressure at engines - - - -	200 lbs. sq. in.
Exhaust back pressure - - - - -	20 lbs. sq. in.
No load steam consumption - -	150 kw. set 2250 lbs./hr.
	50 kw. set 750 lbs./hr.

Running consumption, 30 lbs. steam per kw. generated.

In order to find the steam consumption on which the coal cost must be based, it is necessary to arrive at the daily and annual steam consumption. From this must be deducted such portion of the exhaust steam as is usefully employed for heating and other purposes.

Different conditions apply in summer and in winter owing to the absence of heating load in summer. For the purpose of the following calculation winter and summer are each taken as 182½ days.

Operating Programme—

Winter:

	Daily running hours	Annual running hours	Rated kw.h. per annum	Millions of lbs. of steam per annum
150 kw. set No. 1 -	16	2920	440,000	6·56
150 kw. set No. 2 -	16	2920	440,000	6·56
50 kw. set - -	8	1460	73,000	1·09
Running consumption for units generated:				
350,000 units at 30 lbs. steam per unit				10·5
			953,000	24·71

Average steam consumption per day =

182½

Summer:

	Daily running hours	Annual running hours	Rated kw.h. per annum	Millions of lbs. of steam per annum
150 kw. set No. 1 - -	16	2920	440,000	6·56
150 kw. set No. 2 - -	2	365	55,000	·82
50 kw. set - - -	8	1460	73,000	1·09
Running consumption for units generated:				
184,000 units at 30 lbs. steam per unit				5·44
			568,000	13·91

Average steam consumption per day =

$$\frac{13{\cdot}91 \times 10^6}{182\frac{1}{2}} = \mathbf{76{,}300\ lbs.\ steam.}$$

Live Steam Make-up—The engines are non-lubricated type and exhaust steam is used for all purposes, including laundry and kitchen. The H.W.S. storage is on a 24-hour basis so as to make fullest use of the exhaust. The steam requirements for heating, etc., are as follows:

Winter:

	lbs. steam per day
Cooking - - - - - - -	21,700
H.W.S. - - - - - - -	50,600
Laundry - - - - - - -	38,400
Heating (12,000 lbs. per hr. for 12 hrs. average) - - - - - -	144,000
	254,700
Available exhaust from engines -	135,000
Extra live steam make-up required	119,700

Summer:

	lbs. steam per day
Cooking, H.W.S., Laundry } as above - - - - -	110,700
Available exhaust from engines - -	76,300
Extra live steam make-up required -	34,400

It will be seen from this that neither in summer nor winter will any excess of exhaust steam exist, which would have to be wasted.

Fuel Consumption—The steam consumption attributable to generation is therefore simply the equivalent of the power input to the engines, plus engine losses. The radiation and friction are independent of the units generated, and may be taken as 20 per cent. of the rated kw.h. of the plant running. The electrical losses may be taken as $2\frac{1}{2}$ per cent. of the units generated.

	Winter	*Summer*
Units generated - - - -	350,000	181,000
20% of rated kw.h. of plant running - - - - - -	190,600	113,600
$2\frac{1}{2}$% of units generated - -	8,700	4,500
Total equivalent units generated	549,300	299,100
Heat extracted from total steam passed through engines = equivalent units generated × 3415 B.T.U./unit - -	1875×10^6 B.T.U.'s	1020×10^6 B.T.U.'s
	2895×10^6 B.T.U.'s per annum	

Coal to produce this amount of heat (taking cal. value 12,100 B.T.U.'s/lb. 70 per cent. efficiency of boilers)

$$=\frac{2895 \times 10^6 \times 100}{12,100 \times 2240 \times 70} = 153 \text{ tons.}$$

Annual Running Cost—The total annual running cost may now be arrived at as follows:

Coal as above. 153 tons at 20s. - - - - -	£153
Interest and Depreciation on Plant (including additional cost of boilers) Total cost = £5692 at 10% - - - - - - - -	569
Extra cost of buildings, £1,300: Interest and Depreciation at $6\frac{1}{2}$% - - - - - -	85
Repairs and Maintenance - - - -	60
Extra labour - - - - - - - - -	256
Lubricating Oil and Stores - - - -	45
Insurance £1·5% on £5692 - - - -	85
Total - - - - - -	**£1253**

The cost of public supply offered in this instance was on the basis of a maximum demand charge ranging from £6 per kw. per annum for the first 50 kw. down to £3 10s. od. per kw. above 150 kw. The price per unit was on a similar sliding scale, the first 20 units per kw. per month of maximum demand to be charged at 1d. falling to 0·4d. for all units above 80 per kw. per month; the above being subject to a 'coal clause'.

The annual cost of public supply estimated on the above basis amounted to	**£2441**
The saving due to the combined private generating plant was thus shown to be	**£1188**

Cost per Unit—The Comparative all-in costs per unit are:

Private supply	**0.57d.**
Public supply	**1.10d.**

Another way of looking at the same problem is to assume that the public supply had been taken, in which case the plant would have been practically the same, and the coal consumption would have amounted to 3700 tons per annum, for heating, hot-water supply, cooking, laundry, etc. The adoption of private generation increases this figure to 3853 tons per annum. The savings repay the capital cost in a period of about 6 years after which the additional annual expenditure due to private generation amounts to only £600, giving a cost per unit of 0·27d. for all purposes. At the same time the supply is very reliable and not likely to be interrupted by outside occurrences.

Layout of Plant—Fig. 350 illustrates the general layout of a plant similar to the one discussed above. The installation is a small one initially, with provision for future expansion as indicated.

Plate XXX shows the steam generating plant installed to the authors' designs in accordance with the above details.

RUNNING COST CALCULATION: DIESEL SYSTEM

The figures given for this case are the results of 12 months' working of an existing installation with specially heavy electrical load.

Type of Building	Office Block
No. of occupants	1400

Electrical Consumption—

Lighting	552,000 units p.a.
Heating	148,000
Cooking	124,000
Lifts	98,000
Power (fans, pumps, etc.)	501,000
Total	1,423,000

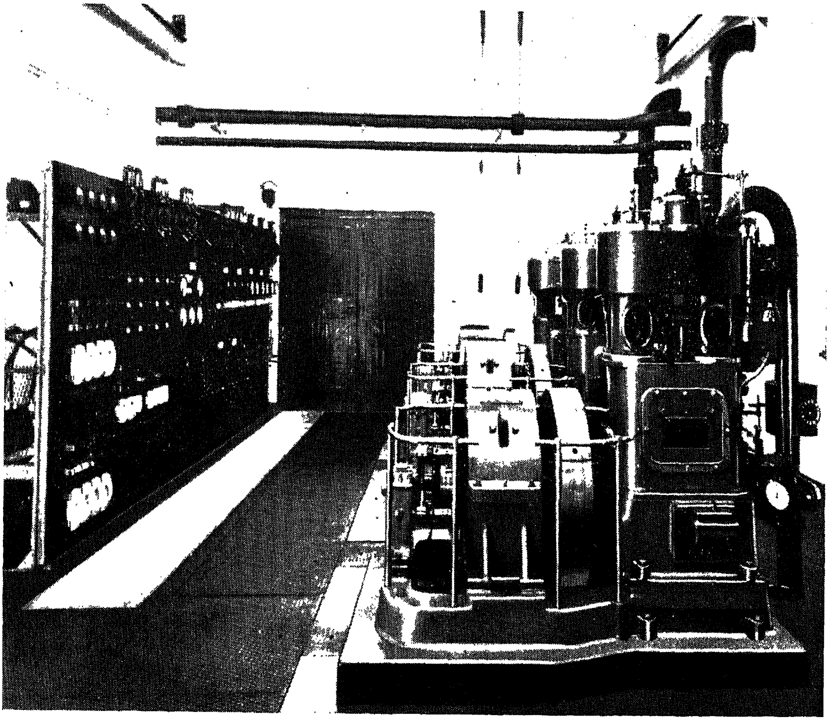


Plate XXX. Steam-driven generating plant installed under the authors' direction at Barrow Hospital

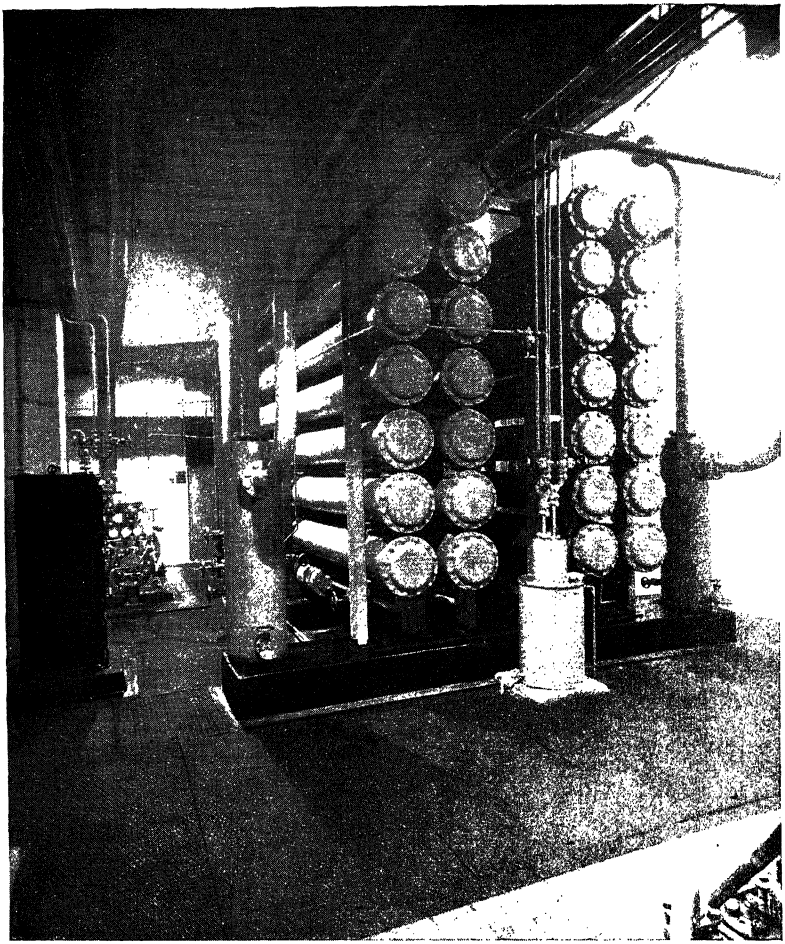


Plate XXXI. Shell and tube ammonia condenser (*see p. 548*)

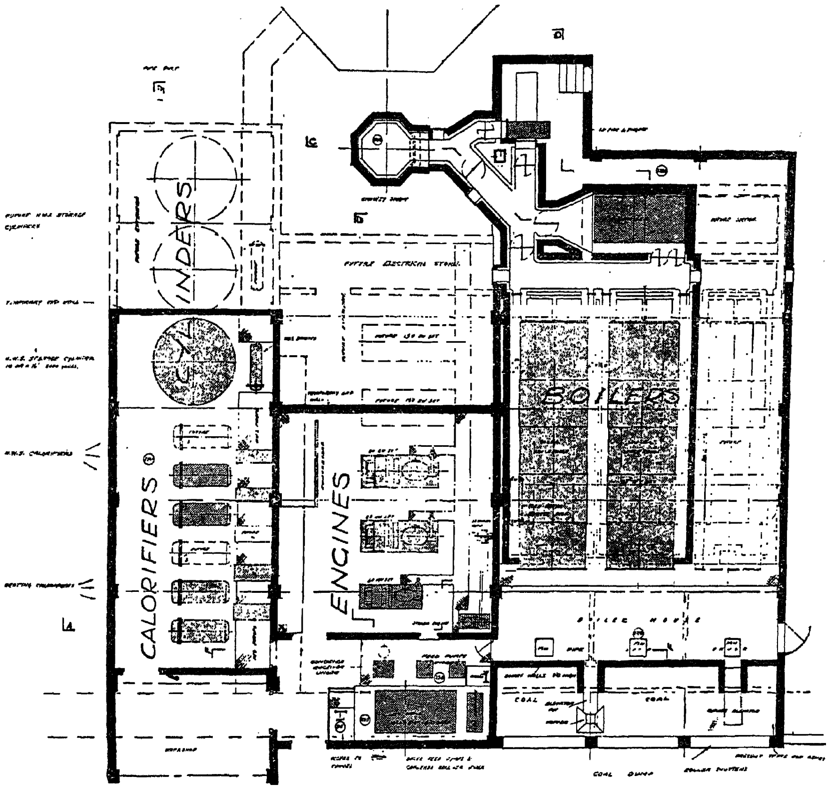


FIG. 350 (*a*).—Plan of a Combined Generating Station at Barrow Hospital. The future plant and buildings are shown by the dotted lines.

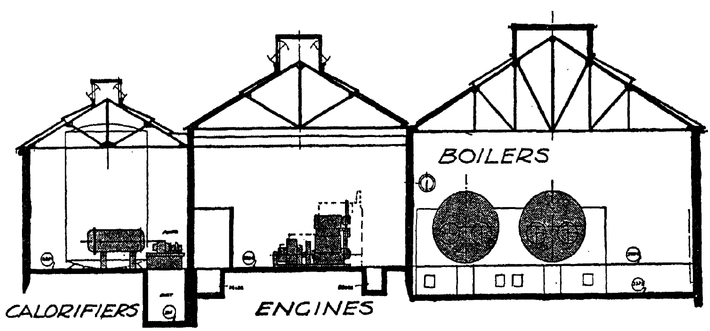


FIG. 350 (*b*).—Cross-section of above.

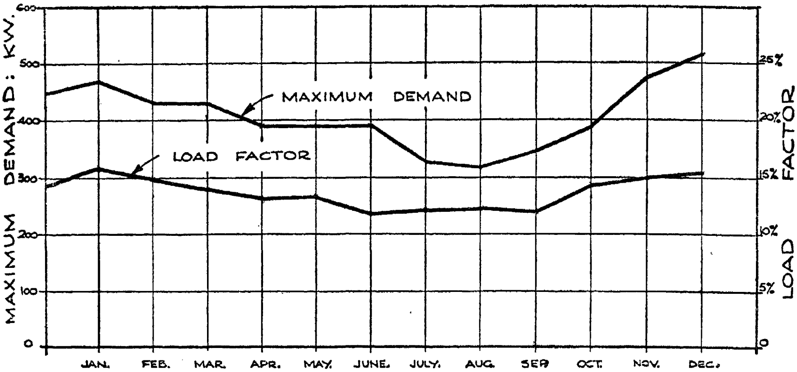


FIG. 351.—Maximum Demand and Load Factor Curves.

Maximum demand, load factor and monthly consumption are indicated on Fig. 248. It will be seen that the annual maximum demand (in December) is 514 kw.

Normal night load - - - - - 80 kw.

Engines Installed—

Four 440 b.h.p. Diesel sets coupled to four 300 kw. generators.

2 sets are running constantly during the day (6 a.m.–8 p.m.) and one during the night.

(When 2 sets running, one down for overhaul and one as standby.)

Annual Running Cost—

Fuel, 486 tons - - - - - -	£1776	
Less equivalent fuel oil supplied to heating and H.W.S. systems, 198 tons -	£728	
		£1048
Interest, £19,000 at 4% - - -		760
Depreciation, average over whole system,		1045
Lubricating Oil, 320 galls. - - -		278
Stores - - - - - - -		89
Insurance and Maintenance - -		513
Labour and Management - - -		1375
		£5108

Cost per Unit—

Total units generated - - - - - -	1,423,000
Less units used in electrical generation - -	62,000
	1,361,000

Total cost per unit = **0.901d.**

The public supply for the same consumption was offered in this special case at the rate of $2\frac{1}{2}$d. per unit, based on a maximum demand charge and unit rate. The saving is therefore 1·6d. per unit, corresponding to £9080 per annum.

The equivalent fuel supplied to the heating and hot-water supply system, for which allowance has been made in the fuel cost, is estimated from the metered water circulated through the engines, and the temperature rise.

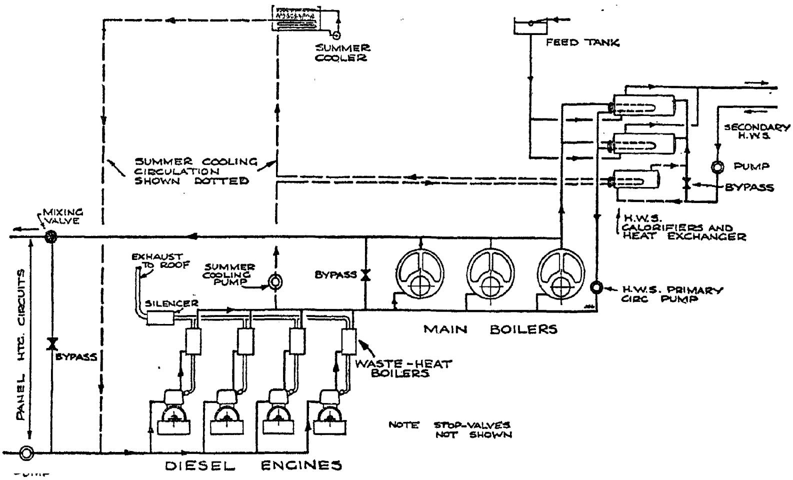


FIG. 352.—Part Piping Diagram of Combined Diesel Generating Station.

Water Circulation—The arrangement of the circulation of this system is shown in Fig. 352. Under winter conditions the circulation is as follows:

(1) The return from the heating panels passes to the circulating pump, which delivers a portion of the water to the engine jackets, from these to the waste-heat boilers, and thence to the heating boilers.

(2) The remaining portion of the return is bypassed, and mixed with the flow from the boiler, so as to give the necessary low temperature outlet for the panel heating system. By this means the cooled water serves for the engine jackets, these requiring the minimum temperature possible, and the make-up of heat over that recovered from the engines plus waste-heat boilers, is provided by the heating boilers.

(3) In summer the waste from the engines and waste-heat boiler passes through a heat exchanger in the hot water supply circuit, and from this to the summer cooler on the roof. A separate pump is provided for this circulation.

(4) The heat exchanger in the hot-water supply system heats the

secondary return water before the latter enters the calorifiers, the primary circulation of which is fed direct at high temperature from the heating boilers. With this arrangement during the summer, for the greater part of the time the heat recovered from the engines furnishes the whole of the hot-water supply requirements.

(5) The summer cooler on the roof disposes of the surplus waste-heat, and consists of a closed circuit cooler over which water is constantly dripped by means of a pump drawing from a shallow tank under the cooler coils.

With the arrangement shown, the same water is constantly recirculated through the engines and waste-heat boilers so that there is no question of these becoming furred up. The jackets are subjected to the full head pressure of the building, but there is no difficulty in arranging for the engines to be suitable for this pressure.

Layout of Plant—Fig. 353 shows the general layout of this plant. The four units are identical, and one is shown in elevation in the lower part of the figure. From this will be seen also the arrangement of the waste-heat boilers, the gases entering at the top and exhausting to a main underneath the engine, thus heating the water on the contra-flow principle. The completed station at the Bank of England, including the four sets mentioned above together with five larger sets of 520 kw. each, is shown in Plate I.

STORAGE BATTERIES

In designing a private generating plant the question arises as to whether a secondary storage battery should be provided. The advantages of a battery are:

(*a*) Avoidance of night running of plant, and of night labour.

(*b*) Security of supply in the event of sudden breakdown of an engine or generator.

(*c*) Steadiness of voltage without close regulation of generator voltage.

The disadvantages of batteries are:

(*a*) Extra capital cost.

(*b*) Extra maintenance cost due to weekly attention to acid level, periodical sludging and re-plating.

(*c*) Low efficiency, the input/output ratio being 75 per cent. at most.

(*d*) Added complication due to necessity for milking booster, control of end-cells, and weekly charging.

(*e*) Extra space occupied.

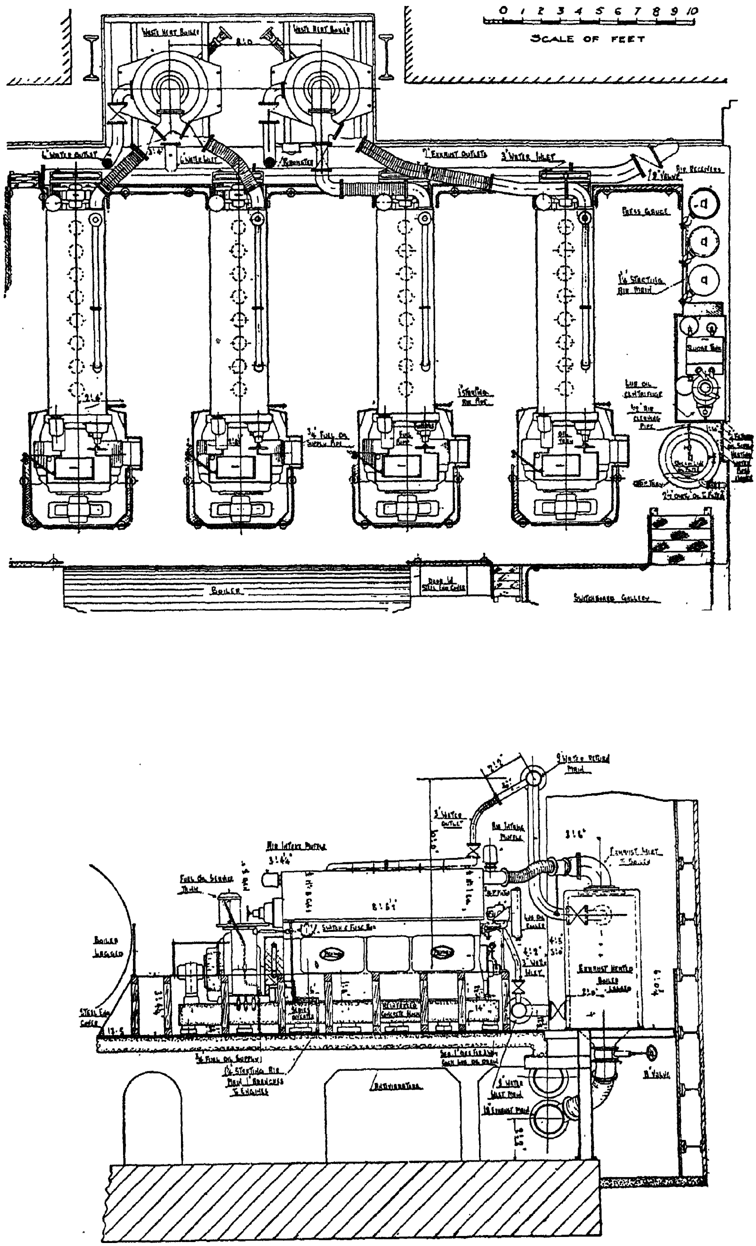


FIG. 353.—Layout of Diesel Engines for Combined Generating Station. The sets shown are 440 h.p.–300 kw. each. Bank of England.
Above: Plan. *Below:* Sectional Elevation of one Engine.

The alternative to a battery is the provision of a small set for continuous night running. In the case of a steam plant this may often be left in charge of the stoker or normal shift engineer without extra labour. In the case of a Diesel plant or where no night labour is provided normally, an extra man or men will be necessary.

The question is largely an economic one, but if all the factors are carefully weighed it will often be found that a battery can be dispensed with and a saving made in overall costs, leading to lower cost of current.

TYPE OF SUPPLY

Where the load is chiefly power, as in a factory, a three phase alternating current supply is usually preferred, owing to the motors and starters for use with this supply being cheaper and requiring less maintenance than those for D.C.

If A.C. is to be used the standard supply, 230/400 volts, 50 cycles should be adopted.

Where lighting, heating and cooking are the main items, as in the case of hospitals, institutions, schools, etc., direct current is equally suitable, and somewhat more simple in the matter of paralleling generators. The supply may then be 200, 220 or 240 volts two wire, or 400, 440 or 480 three wire. If the latter, rotary balancers will be necessary to deal with out-of-balance loads. It will be found that a three wire system of distribution requires about one-third of the copper of a two wire for equal load and voltage drop, and this saving, and the saving on high voltage motors, must be weighed against the cost of the balancers.

INDEX

Where a subject occupies several consecutive pages, the first page number only is given. References thus: '370-8', indicate scattered references in the pages given.

References are not given to items in tables. Thus for properties of corkboard, for instance, see under 'Insulation', 'Conductivity', etc.

The following abbreviations are used:

A.-c.	Air-conditioning.
Elec.	Electric, Electrical.
Htg.	Heating.
H.w.s.	Hot water supply.
H.P.H.W.	High pressure hot water.
Stm.	Steam.
Th.-c.	Thermostatic control.
Th. st.	Thermal storage.
Refr.	Refrigeration.
Ventn.	Ventilation or Ventilating
U.H.	Unit heater.

Where one of the page references is to a chapter or section on the subject, this reference is put first, even if not in numerical page order.

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